ENERGY
MANAGEMENT
SERIES

11

FOR INDUSTRY
COMMERCE
AND INSTITUTIONS

Refrigeration and Heat Pumps

Energy, Mines and Resources Canada   Énergie, Mines et Ressources Canada

Canada

PREFACE

Much has been learned about the art and science of managing energy during the past decade. Today, energy management is a seriously applied discipline within the management process of most successful companies.

Initially, in the early 1970's, energy conservation programs were established to alleviate threatened shortages and Canada's dependency on off-shore oil supplies. However, dramatic price increases quickly added a new meaning to the term "energy conservation" — reduce energy costs!

Many industrial, commercial and institutional organizations met the challenge and reduced energy costs by up to 50%. Improved energy use efficiency was achieved by such steps as employee awareness programs, improved maintenance procedures, by simply eliminating waste, as well as by undertaking projects to upgrade or improve facilities and equipment.

In order to obtain additional energy savings at this juncture a greater knowledge and understanding of technical theory and its application is required in addition to energy efficiency equipment itself.

At the request of the Canadian Industry Program for Energy Conservation, the Commercial and Institutional Task Force Program and related trade associations, the Industrial Energy Division of the Department of Energy, Mines and Resources Canada, has prepared a series of energy management and technical manuals.

The purpose of these manuals is to help managers and operating personnel recognize energy management opportunities within their organizations. They provide the practitioner with mathematical equations, general information on proven techniques and technology, together with examples on how to save energy.

Further information concerning the manuals listed below or regarding material used at seminars/workshops including actual case studies, please write to:

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INTRODUCTION

Throughout history, humans have used various forms of refrigeration. Simple cooling arrangements, such as those provided by ice boxes and root cellars, allowed long term storage of perishable foods. These, and other simple techniques, though largely supplanted by mechanical refrigeration equipment, are still used by campers, cottagers and people in remote or less developed areas.

Mechanical refrigeration systems were first built in the late nineteenth century, but did not become commonplace until the 1940s. Although mechanical refrigeration provides benefits such as refrigerated storage independent of season or climate, and better living and working environments, the energy costs related to operation of these systems are significant. This module examines refrigeration and heat pump systems and identifies where energy consumption and costs may be reduced.

Purpose

The following summarizes the purpose of this module.

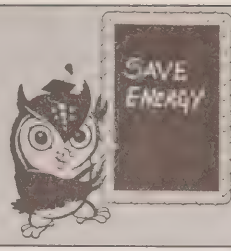
- Introduce the subject of Refrigeration and Heat Pumps as used in the Industrial, Commercial and Institutional Sectors.
- Make building owners and operators aware of the potential energy and cost savings available through the implementation of Energy Management Opportunities.
- Provide methods of calculating the potential energy and cost savings, using worked examples.
- Provide a set of worksheets that can be used to perform calculations for existing and/or proposed systems to establish energy and cost savings potential.

Contents

The contents have been subdivided into the following sections to describe the concepts, purposes, and uses of refrigeration and heat pumps.

- *Fundamentals* of refrigeration and heat pumps, with examples where necessary, to provide a basic understanding of the concepts used to develop the equations and calculations.
- *Equipment/Systems* describes typical refrigeration and heat pump equipment used in the Industrial, Commercial and Institutional sectors.
- *Energy Management Opportunities* are described and supported by estimated figures for energy and cost savings and simple payback calculations.
- Appendices include a glossary, tables, conversion factors and blank worksheets.

FUNDAMENTALS



Refrigeration and heat pump systems are used to transfer heat energy from, or to, products, processes or buildings. Most everyone is familiar with a domestic refrigerator. Energy in the form of electricity is used to power mechanical equipment designed to transfer heat from a colder, low energy level to a warmer, high energy level. This energy input must be carefully managed to minimize cost especially in larger systems encountered in Industrial, Commercial and Institutional facilities. Numerous opportunities exist for improving energy management and reducing operating costs.

Building and Process Loads

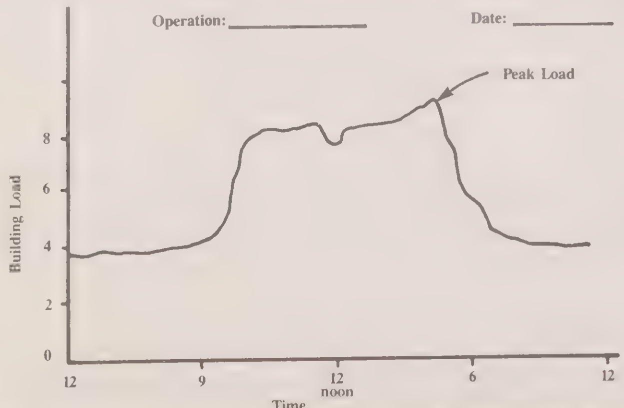
The amount of refrigeration capacity required for a specific cooling application is affected by a number of factors, whether for comfort air-conditioning or a process application. For example, building refrigeration requirements, or loads, are dependent on:

- Building construction (walls, roof, windows, doors).
- Ventilation air (amount of outside air, exhaust air).
- Internal load (lights, people, production equipment).
- Building function (factory, warehouse, office building, recreation centre, health care centre, school).
- Weather (summer, winter).
- Infiltration (leaks around doors, windows and stack effect).

Process refrigeration loads, while influenced to a certain degree by these same factors, are dictated mainly by the production requirements unless thermal storage techniques are used. Generally, the process loads are more predictable than building loads both in magnitude and scheduling.

When a refrigeration load is plotted against time, a "load profile" is generated (Figure 1). The profile shows the load of a particular system or a number of systems in an entire facility at various times in a day. This information enables a building owner or operator to determine the magnitude and time of occurrence of "peaks", or periods of highest demand, as well as total loads. The peaks on a typical load profile highlight areas where the most attractive Energy Management Opportunities exist.

Often, a load profile will show the electrical power requirements rather than the actual refrigeration loads. By careful management of the refrigeration loads, the electrical peaks can be reduced. Refer to Module 3, Electrical for more information on the effects of "peak shaving".



Typical Heating Or Cooling Load Profile
Figure 1

Refrigeration - Terms and Definitions

Refrigeration is primarily used for cooling a product, liquid or gas. Frequently, an intermediate step is involved to transfer heat or cooling energy to, or from, another process. The production of chilled water for process cooling and air-conditioning systems is a common example.

Certain concepts of energy, temperature, work, and their interrelationships must be understood before investigating Energy Management Opportunities. The *heat pump* section of this module describes how refrigeration can be used for heating.

Refrigerant

A refrigerant is a fluid used for transferring heat in a system, where the fluid absorbs heat at a low temperature and pressure, and rejects heat at a higher temperature and pressure. Usually this heat transfer involves a change of state of the refrigerant fluid from liquid to gas and vice versa.

Temperature

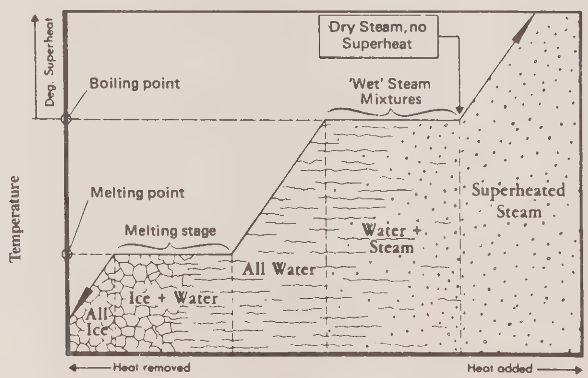
Temperature is an indication of the heat energy stored in a substance. When the temperature of a substance is decreased to −273°C or 0 K (Kelvin), known as *absolute zero*, the substance contains no heat energy and all molecular movement stops.

Pressure

Pressure is the force exerted on a surface, per unit area, and is expressed in kilopascals (kPa) or megapascals (MPa).

Changes of State

When sufficient heat is added or removed, most substances undergo a change of state. The temperature remains constant until the change of state is complete. Change of state can be from solid to liquid, liquid to vapor or vice versa. Typical examples are ice melting and water boiling (Figure 2).



Example Of Change Of State
Figure 2

Energy in Liquids and Vapors

When a liquid is heated, the temperature of the liquid rises to the *boiling point*. This is the highest temperature to which the liquid can rise at the measured pressure.

The heat absorbed by the liquid in raising the temperature to the boiling point is called *sensible heat*. The heat required to convert the liquid to vapor at the same temperature and pressure is called *latent heat*.

Heat Transfer

Heat energy can flow only from a higher to a lower temperature level unless energy is added to reverse the process. Heat transfer will occur when a temperature difference exists *within a medium* or *between different media*. Higher heat transfer rates occur at higher temperature differences.

Latent Heat of Fusion

For most pure substances there is a specific melting/freezing temperature, relatively independent of the pressure. For example, ice begins to melt at 0°C. The amount of heat necessary to melt one kilogram of ice at 0°C to one kilogram of water at 0°C is called the latent heat of fusion of water and equals 334.92 kJ/kg. The removal of the same amount of heat from one kilogram of water at 0°C will change it back to ice.

Evaporation and Condensation

Unlike freezing and melting, evaporation and condensation can take place at almost any temperature and pressure combination. *Evaporation* is the gaseous escape of molecules from the surface of a liquid and is accomplished by the absorption of a considerable quantity of heat without any change in temperature. The vapor that leaves the surface of a boiling liquid is called *saturated vapor*. The quantity of heat required to make the change of state is called the *latent heat of vaporization*. *Condensation* occurs when the gaseous molecules return to the liquid state.

Liquids, including refrigerants, evaporate at all temperatures with increased rates of evaporation taking place at higher temperatures. The evaporated gases exert a pressure called the vapor pressure. As the temperature of the liquid rises, there is a greater loss of the liquid from the surface which increases the vapor pressure. *Boiling* occurs when the vapor pressure reaches the pressure of the surrounding space. During boiling, vapor is generated at a pressure equal to the gas pressure on the surface. If the pressure on the surface is increased, boiling takes place at a higher temperature and the boiling point is said to increase. Similarly, a reduction in the pressure will lower the boiling point.

Quality of Vapor

Theoretically, when vapor leaves the surface of a liquid, it is pure and saturated at the particular temperature and pressure. In actual practice, tiny liquid droplets escape with the vapor. When a mixture of liquid and vapor exists, the ratio of the mass of the liquid to the total mass of the liquid and vapor mixture is called the "quality", and is expressed as a percentage or decimal fraction.

Process

A process is a physical or chemical change in the *properties* of matter, or the conversion of energy from one form to another. In refrigeration, a process is generally defined by the condition (or properties) of the refrigerant at the beginning and end of the process.

Cycle

A cycle is a series of processes where the end point conditions or properties of the substance are identical to the initial conditions. In refrigeration, the processes required to produce a cooling effect are arranged to operate in a cyclic manner so that the refrigerant can be reused.

Work

Work is the energy which is transferred by a difference in pressure or force of any kind. Work is subdivided into shaft work and flow work.

Shaft work is mechanical energy used to drive a mechanism such as a pump, compressor or turbine. *Flow work* is the energy transferred into a system by fluid flowing into, or out of, the system. Both forms of work are expressed in kilojoules, or on a mass basis, kJ/kg.

Refrigerant Tables

Common properties of refrigerants are tabulated for both liquid and vapor phases, and at different temperature-pressure conditions.

Saturation Temperature

Saturation temperature, normally the first column in a refrigerant table, and given in K, is the temperature at which boiling will occur to produce vapor at the given saturation pressure. For example, Table 1 shows that refrigerant R-12 will boil at 228K (−45°C) when at an absolute pressure of 0.050035 MPa. The same table shows that R-12 will boil at 260K (−13°C) if the pressure is increased to 0.19566 MPa.

Saturation Pressure

Saturation pressure is normally the second column in a refrigerant table and is expressed as MPa(absolute). To obtain gauge pressure subtract 0.10132 MPa (101.32 kPa) from the absolute pressure.

Density of Saturated Liquid

The density of liquid at saturation temperature and pressure is expressed in kg/m^3. The specific volume of the refrigerant liquid can be calculated by taking the inverse of the density.

$$\text{Specific Volume} = \frac{1}{\text{Density}}$$

Specific Volume of Saturated Vapor

The specific volume of saturated vapor is the volume occupied by one kilogram of dry saturated gas at the corresponding saturation temperature and pressure, and is expressed in m^3/kg. Density of the vapor can be calculated by taking the inverse of the specific volume.

$$\text{Density} = \frac{1}{\text{Specific Volume}}$$

Enthalpy

The total energy contained in a refrigerant is called the enthalpy. Most refrigerant tables assume, for convenience of calculations, that the saturated liquid at -40°C has zero energy.

- *Enthalpy of liquid* (h_f) is the amount of energy contained in one kilogram of the liquid at a particular temperature, and is expressed in kJ/kg.
- *Enthalpy of vapor* (h_g) is the total energy contained in dry saturated vapor at a particular temperature and saturation pressure, and is expressed in kJ/kg.
- *Latent heat of vaporization* (h_{fg}) is the amount of energy required to evaporate one kilogram of liquid at a given temperature and pressure and is the difference between the enthalpy of the liquid and the vapor. It is expressed in kJ/kg.

The enthalpy equation is: $h_{fg} = h_g - h_f$

- *Enthalpy of a mixture* is a value necessary in the calculation of most practical applications because a refrigerant usually contains a mixture of both vapor and liquid. If the quality of the vapor is "x", then :

$$h = h_f + x\,(h_g - h_f)$$

Where, h = Enthalpy of "wet" vapor (kJ/kg)

h_f = Enthalpy of the liquid (kJ/kg)

h_g = Enthalpy of the vapor (kJ/kg)

x = Quality of the vapor (decimal fraction)

Entropy (s)

Entropy can be described as a measure of the molecular disorder of a substance, and is used to describe the refrigeration cycle.

- *Entropy of saturated liquid* (s_f) at a given temperature and pressure condition is expressed in kJ/(kg.K).
- *Entropy of saturated vapor* (s_g) at a given temperature and pressure condition is expressed in kJ/(kg.K).
- *Entropy of vaporization* (s_{fg}), is the difference in entropy between the saturated liquid and the saturated vapor.

Superheated Vapor

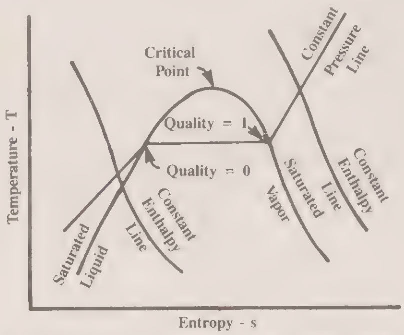
Superheated vapor is saturated vapor to which additional heat has been added, raising the temperature above the boiling point. A superheated condition can also be obtained by reducing, at constant temperature, the pressure of the vapor.

Superheated vapor is free of all liquid and the quantity of superheat is expressed in *degrees of superheat*. This is the number of degrees Celsius to which the vapor is heated above the saturation temperature. The value of any property of a superheated vapor corresponding to the temperature and pressure can be read directly from tables. See Table 2 for refrigerant R-12 superheat data.

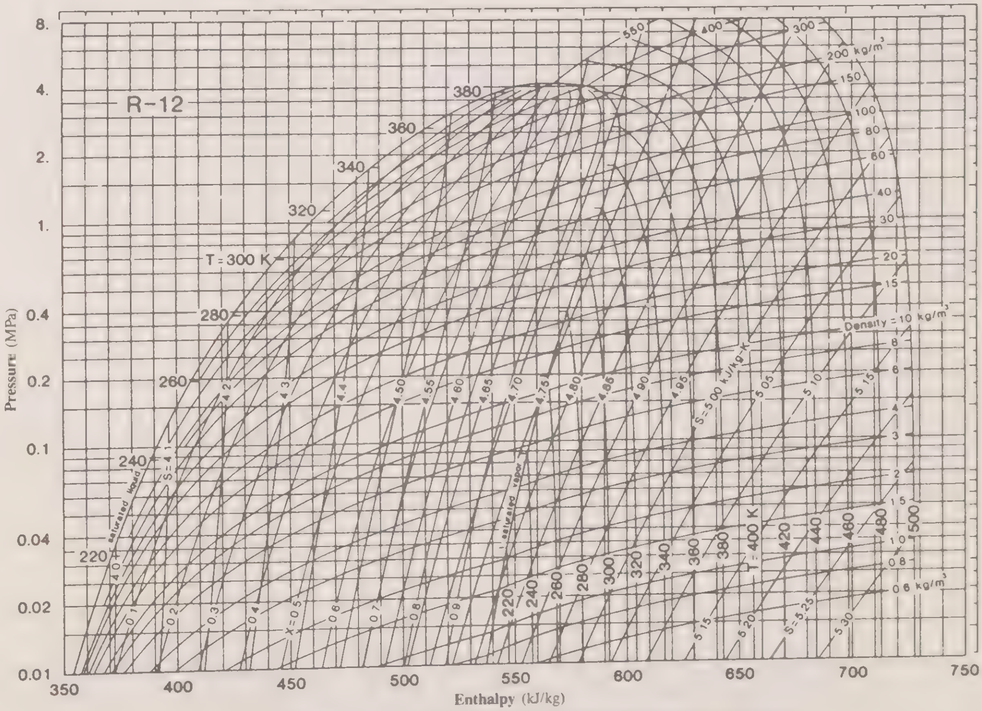
Superheated gases are not practical for refrigeration applications because larger equipment is required as the quantity of superheat increases. It is usual to *desuperheat* refrigerant gas that gets superheated during the process, to reduce the volume of gas to be handled by the refrigeration equipment.

Property Diagrams

Property diagrams are graphical summaries of the main properties listed in the refrigerant tables. Changes in the properties can be plotted to assist in visualizing the processes that cause the changes. Two common versions are the Mollier and T-s diagrams (Figure 3).



T-s Diagram



Pressure-Enthalpy Diagram For Refrigerant 12 (Mollier)

Figure 3

Courtesy of ASHRAE

The *Mollier diagram* presents the pressure in MPa on the vertical axis, and the enthalpy in kJ/kg on the horizontal axis. Temperature in K, density and entropy are shown as curves.

A simplified *T-s diagram* will be used in this module, as it provides an easier tool for understanding the refrigeration cycle. Temperature is plotted on the vertical axis, and entropy on the horizontal axis. Pressure and enthalpy are shown as curves on the diagram.

Both diagrams have one prominent feature, the *saturation line*. This bell-shaped curve represents the boundary between vapor and liquid states. The left side of the curve is the *saturated liquid* line, the right side is the *saturated vapor* line. The area under the curve is the saturation region, where vapor and liquid coexist. The quality of the fluid is 0 at the left boundary, 1 (or 100 per cent) at the right boundary. The peak in the curve is the *critical point* where the difference between liquid and vapor become indistinct.

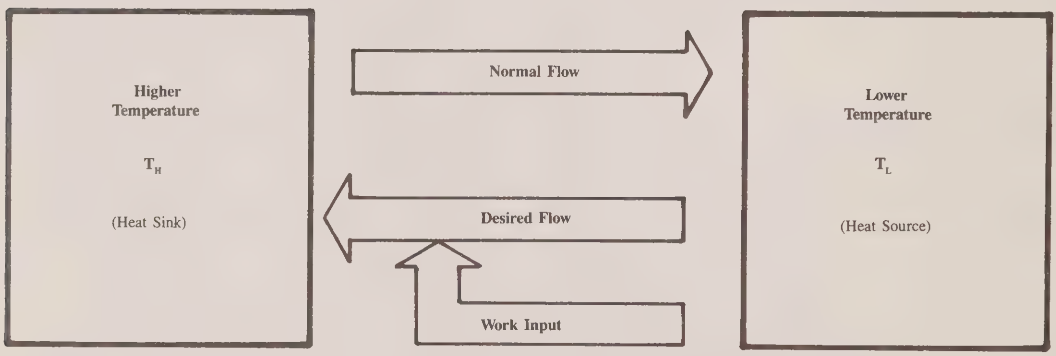
Basic Refrigeration Cycle

The main purpose of a refrigeration system is to remove heat energy from a low temperature medium (i.e. the *heat source*), and transfer this heat energy to a higher temperature medium (i.e. the *heat sink*). There are many examples of refrigeration.

- Cooling air for air-conditioning.
- Chilling liquid for process cooling such as in the food industry.
- Chilling brine in order to freeze a sheet of ice (e.g. a hockey arena).

In certain cases, the primary use of the refrigeration equipment is to heat the high temperature medium to a higher level. In this case, the refrigeration system is called a *heat pump*.

Much like water which flows downhill because of gravity, energy (heat) always flows from a higher to a lower energy level. Moving heat from a lower to a higher temperature level is contrary to the "natural" energy flow pattern, and requires the input of *work* (Figure 4).



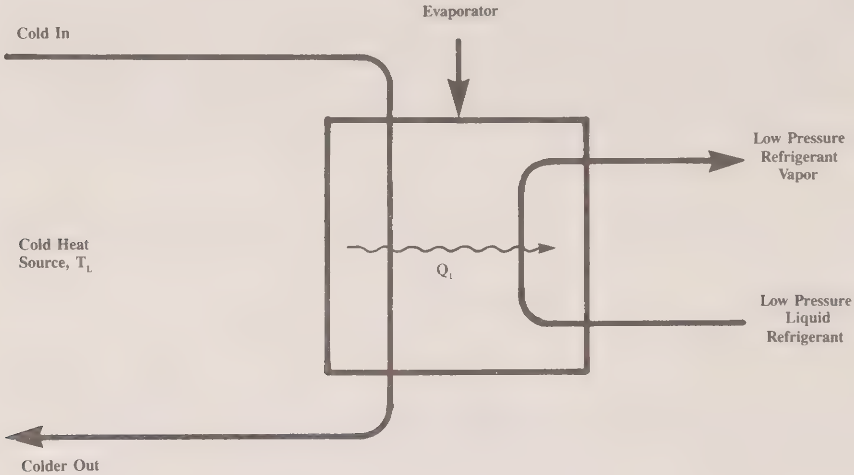
Heat Flow Patterns
Figure 4

A heat transfer fluid, called a refrigerant, is used to transfer the heat energy. Initially, the refrigerant, because it is at a temperature lower than the heat source, absorbs heat. The temperature of the refrigerant is increased during the process to a temperature higher than the heat sink. The refrigerant therefore rejects the heat.

Four basic steps are required to complete the refrigeration cycle.

Step One: Evaporation

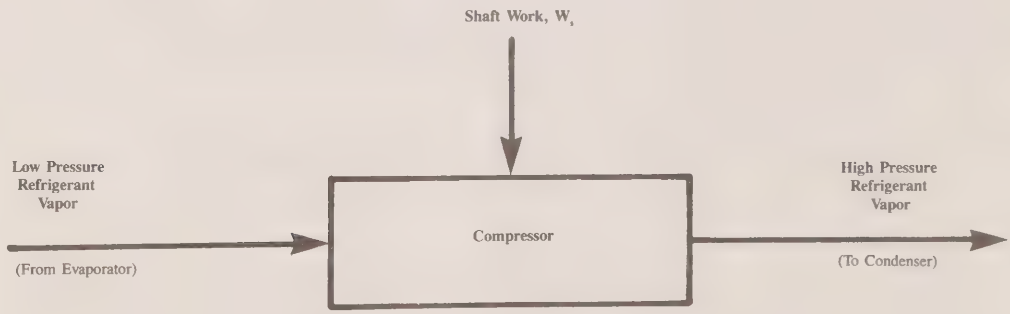
A cool, low-pressure liquid refrigerant is brought into contact with the heat source, the medium to be cooled. The refrigerant, being at low pressure, absorbs heat and boils, producing a low-pressure vapor at the saturation condition (Figure 5). The heat exchanger used for this process is called the *evaporator*.



Evaporation
Figure 5

Step Two: Compression

The addition of shaft work by a compressor raises the pressure of the refrigerant vapor. The addition of heat may also be used to raise the pressure. Increasing the gas pressure raises the boiling and condensing temperature of the refrigerant. Once the refrigerant gas has been sufficiently compressed, its boiling point will be above the temperature of the heat sink, the higher temperature medium (Figure 6).

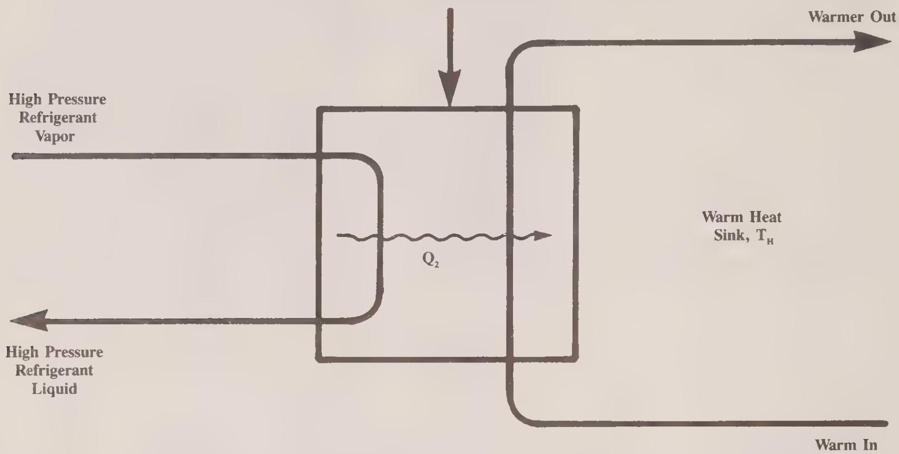


Compression
Figure 6

Step Three: Condensing

The high-pressure refrigerant gas now carrying the heat energy absorbed at the evaporator, and the work energy from the compressor, is pumped to a second heat exchanger called the *condenser*. Because the refrigerant's condensing temperature is higher than that of the heat sink, heat transfer will take place, condensing the refrigerant from a high-pressure vapor to a high-pressure saturated liquid. The heat source has been cooled by "pumping" the heat energy to the heat sink (Figure 7).

Instead of using a condenser to reject the heat, the refrigerant vapor could be discharged to atmosphere, but this approach is impractical. Condensing the gas allows reuse at the beginning of the next cycle. In practice, the condenser cools the refrigerant further, *subcooling* it below the condensing temperature. Subcooling of 11°C is usually provided at the condenser to reduce flashing when the refrigerant pressure is reduced at the throttling device. The subcooling reduces the amount of gas entering the evaporator, improving the system performance.



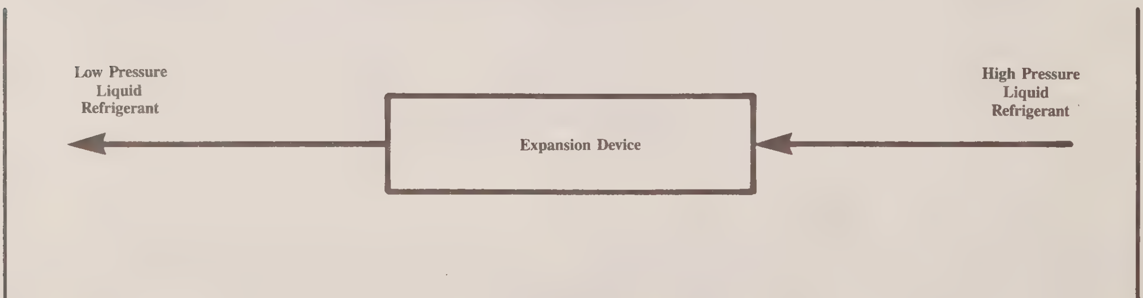
Condensing
Figure 7

Step Four: Expansion

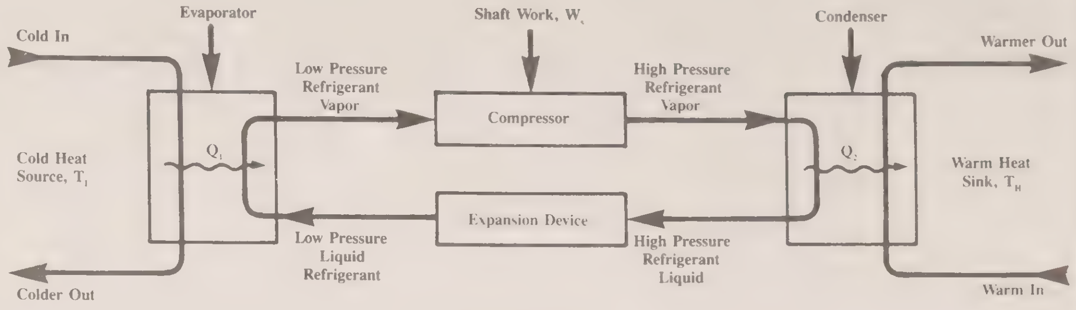
The condensed liquid is returned to the beginning of the next cycle. Its pressure must be reduced to prevent the high-pressure liquid from entering the low pressure evaporator, and to reduce the boiling temperature of the refrigerant to below the temperature of the heat source. A throttling device such as a *valve, orifice plate or capillary* is generally used for this purpose. Energy lost through this reduction of pressure must be offset by additional energy input at the pressurization stage (Figure 8).

All Steps Combined: The Complete Cycle

The basic refrigeration cycle, with all steps combined, is shown diagrammatically in Figure 9.



Expansion
Figure 8



Basic Refrigeration Cycle
Figure 9

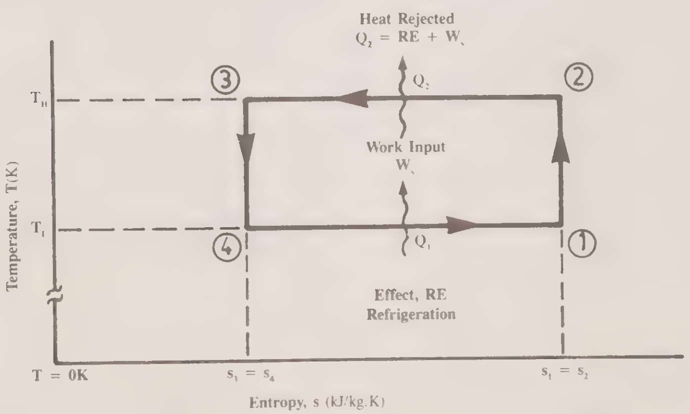
Reverse Carnot Cycle

The Carnot Cycle is a theoretical model representing the basic processes of a refrigeration cycle. It is a model *heat engine* where the addition of heat energy to the engine will produce work. The Reverse Carnot cycle transfers heat when work is applied to the engine. From the model, the maximum theoretical performance can be calculated, establishing criteria to which real refrigeration cycles can be compared.

The following processes occur in the Reverse Carnot Cycle (Figure 10).

- 4 to 1 is the absorption of heat at the evaporator, a constant temperature boiling process at T_L.
- 1 to 2 is constant entropy (ideal) compression. Work input is required and the temperature of the refrigerant increases.
- 2 to 3 is heat rejection at the condenser, a constant temperature process at T_H.
- 3 to 4 is constant entropy (ideal) expansion from a higher to a lower pressure through the throttling device.

From the diagram, the concept of *Coefficient of Performance* (COP) is derived. The COP is the ratio of the cooling or *Refrigeration Effect* (RE), to the work required to produce the effect.



Reverse Carnot Cycle
Figure 10

Coefficient of Performance

The refrigeration effect is represented as the area under the process line 4-1.

$$RE = T_L \times (s_1 - s_4)$$

Where, RE = Refrigeration effect (kJ)

T_L = Temperature (K)

s_1, s_4 = Entropy [kJ/(kg.K)]

The theoretical work input (W_S) (i.e. energy requirement) for the cycle is represented by the area "within" the cycle line 1-2-3-4-1.

$$W_S = (T_H - T_L) \times (s_4 - s_1) \text{ kJ/kg}$$

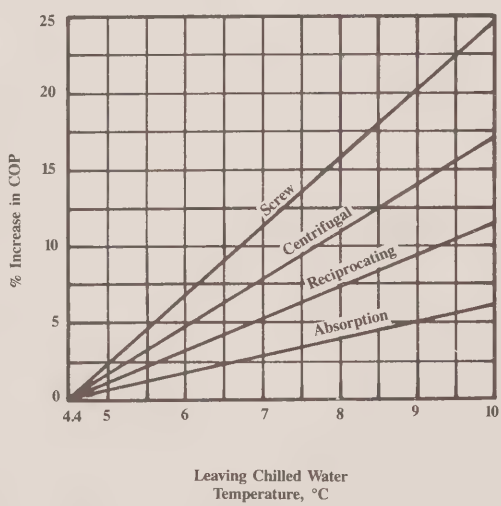
The equation for coefficient of Performance (COP) is obtained by dividing the refrigeration effect (RE) by the theoretical work input (W_s).

$$COP = \frac{RE}{W_S} = \frac{T_L \times (s_1 - s_4)}{(T_H - T_L) \times (s_1 - s_4)}$$

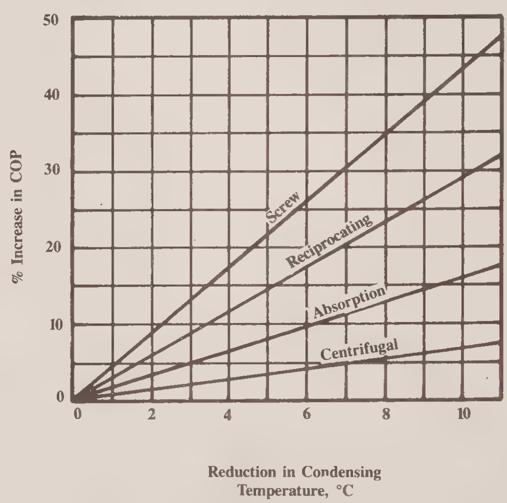
The coefficient of performance for this theoretical system is temperature dependent and can be reduced to:

$$COP(Ideal) = \frac{T_L}{(T_H - T_L)}$$

Actual systems are not as efficient as the ideal or theoretical model (i.e. lower COP), but the following general conclusion applies: *The smaller the temperature difference between the heat sink and the heat source, ($T_H - T_L$) the greater the efficiency of the refrigeration (or heat pump) system*. The COP, a measure of the energy required to produce a given refrigeration effect, is an excellent means of comparing the efficiencies of similar equipment.



Effect Of Condensing Temperature On Chiller COP
Figure 11



Effect Of Evaporator Temperature On Chiller COP
Figure 12

Example:Two refrigeration machines of similar capacity are compared. One has a COP of 4.0 while the second has a COP of 3.0 at the same operating conditions. The first machine with the higher COP is the most efficient, producing 1.33 times the refrigeration effect for the same work input of the second machine. Figures 11 and 12 show the effect of evaporator and condenser temperatures on the COP for various types of chillers.

The theoretical COP can also be expressed in terms of enthalpy, where the difference in energy content of the refrigerant at various points of the cycle define the cooling effect and the work input.

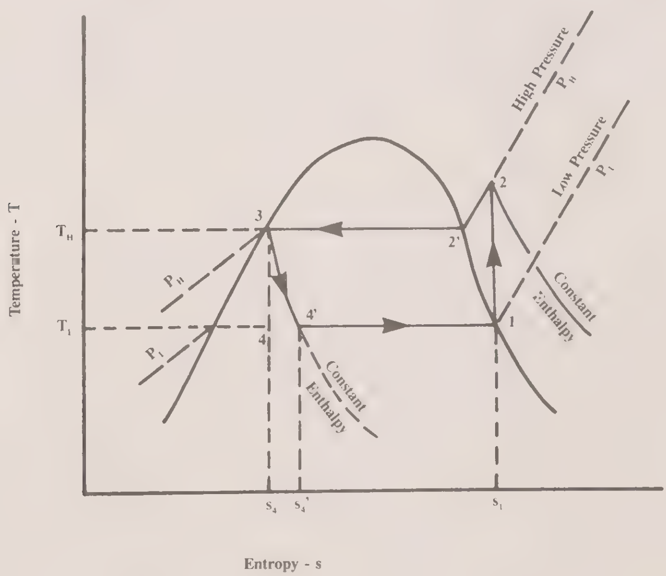
$$COP = \frac{(h_1 - h_4)}{(h_2 - h_1)}$$

Theoretical Vapor Compression Cycle

The Carnot cycle, although a useful model to assist in the understanding of the refrigeration process, has certain limitations. One limitation is the lack of accounting for changes of state. Figure 13 shows a vapor compression cycle approximating the effect of the cycle on the refrigerant, assuming ideal equipment, where:

- 1 - 2 Compression.
- 2 - 2' Desuperheating.
- 2' - 3 Constant Temperature Condensation.
- 3 - 4' Throttling.
- 4' - 1 Constant Temperature Evaporation.

Assuming that the compression process starts at point 1 as a saturated vapor, energy added in the form of shaft work will raise the temperature and pressure. Ideally, this is a constant entropy process represented by a vertical line on the T-s diagram. The net result is superheating of the vapor to point 2. Process 2-2'-3 is heat rejection at the condenser. Step 2-2' is the initial desuperheating of the hot gas at the condenser or intermediate equipment, and 2'-3 is the condensation process.



Basic Refrigeration Cycle
Figure 13

Coefficient of Performance

As in the Reverse Carnot cycle, the coefficient of performance is:

$$\text{COP (Refrig)} = \frac{\text{refrigeration effect}}{\text{work input}}$$

$$= \frac{T_L}{(T_H - T_L)}$$

$$= \frac{h_1 - h_4'}{h_2 - h_1}$$

Where, $h_4' = h_3$

Departures from the ideal Carnot cycle are apparent.

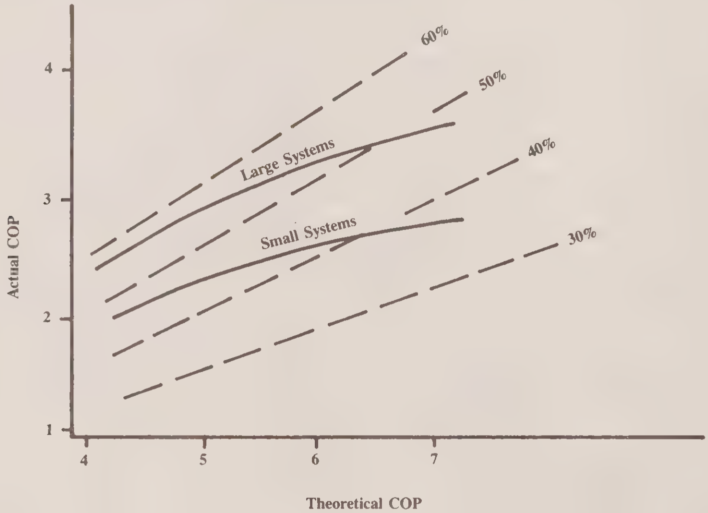
- $[h_2 - h_1]$(theoretical) is larger than $[h_2 - h_1]$(Carnot).
- $[h_1 - h_4']$(theoretical) is smaller than $[h_1 - h_4]$(Carnot).

The net effect is a COP reduction.

The throttling process reduces the refrigerant pressure from the condensing (high) pressure side to the evaporator (low) pressure side. By definition, throttling is a *constant enthalpy* process. The enthalpy at point 3 is equal to that at point 4', thus $h_3 = h_4'$. Energy is degraded in this process, therefore the entropy must increase from point 3 to 4'.

Practical Considerations

Refrigeration and heat pump cycles are more complex than the theoretical vapor compression cycle discussed in this module. Practical limitations such as equipment size, system pressure, and design temperatures at the evaporator and condenser, reduce the effectiveness of actual systems. Actual COPs are 20 to 30 per cent of the theoretical COP based on the Carnot cycle operating at the same conditions. Individual components, such as the compressor, may have an effectiveness of 40 to 60 per cent of the theoretical COP (Figure 14). These limitations, and techniques used to reduce their input on cycle efficiency, are now discussed.

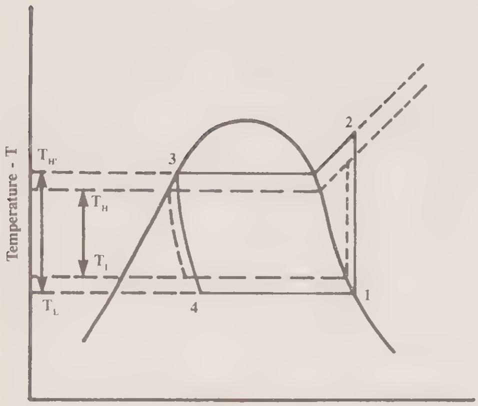


Effectiveness Of Reciprocating Compressors
Figure 14

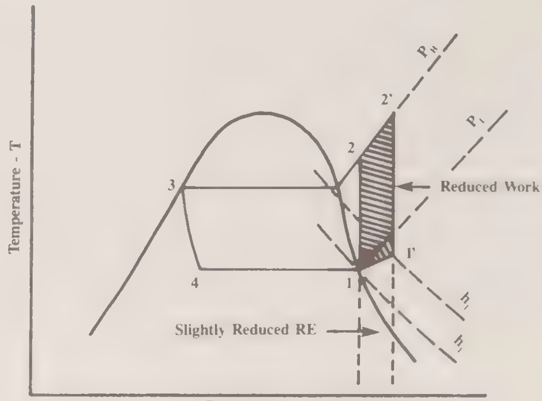
Heat Transfer

Operating temperatures in actual cycles are established to suit the temperatures required at the cold medium and the temperature acceptable for the heat sink. The practical temperature gradient required to transfer heat from one fluid to another through a *heat exchanger* is in the range of 5 to 8 °C. This means that the refrigerant entering the evaporator should be 5 to 8°C colder than the desired medium temperature. The saturation temperature at the condenser should be 5 to 8 °C above the temperature of the heat rejection medium (Figure 15).

The area enclosed by line 1-2-3-4'-1, which describes the cycle, has increased because of the temperature difference required to drive the transfer process. There has been an increase in the work required to produce the refrigeration effect because the temperature difference has increased, ($T_H - T_L$).



Heat Exchanger Limitations
Figure 15



Effect Of Desuperheating
Figure 16

Superheat

In the refrigerant cycle, refrigerant gas becomes superheated at the evaporator and at the compressor (Figure 16).

During the evaporation process the refrigerant is completely vaporized part-way through the evaporator. As the cool refrigerant vapor continues through the evaporator, additional heat is absorbed which superheats the vapor. Pressure losses, caused by friction, further increase the amount of superheat. When the superheating occurs at the evaporator, the enthalpy of the refrigerant is raised, extracting additional heat and increasing the refrigeration effect of the evaporator. When superheating occurs in the compressor suction piping, no useful cooling occurs.

The increase in refrigeration effect, caused by superheating in the evaporator, usually is offset by a decrease in refrigeration effect at the compressor. Because the volumetric flow rate of a compressor is constant, the mass flow rate and refrigerating effect are reduced by decreases in refrigerant density caused by the superheating. The relative effects of increases in enthalpy and decreases in density must be calculated in detail. A study of the system design may be practical only for systems over 500 kW in capacity. There is a loss in refrigerating capacity of about one per cent for every 2.5°C of superheat in the suction line of a reciprocating compressor. Insulation on suction lines will minimize the undesirable heat gain.

Refrigerant superheating also occurs at the compressor. The refrigerant enters the compressor as a saturated vapor. Increasing the pressure will increase the temperature and cause superheat. Friction, system inefficiency and the work added raise the entropy and superheat above that occurring in the theoretical cycle. Superheat, caused by the compression process, does not improve cycle efficiency, but results in larger condensing equipment and large compressor discharge piping.

Desuperheating is the process of removing excess heat from superheated refrigerant vapor, and when accomplished by means *external to the cycle*, can be beneficial to system performance. Desuperheating the suction gas is often impractical because of the low temperatures (less than 10 °C) and the small amount of available energy . Some superheat is required to prevent slugs of liquid refrigerant from reaching the compressor and causing serious damage.

At design conditions, superheat can account for 20 per cent of the heat rejected at the condensers, and often raises condensing temperatures above 45 °C.

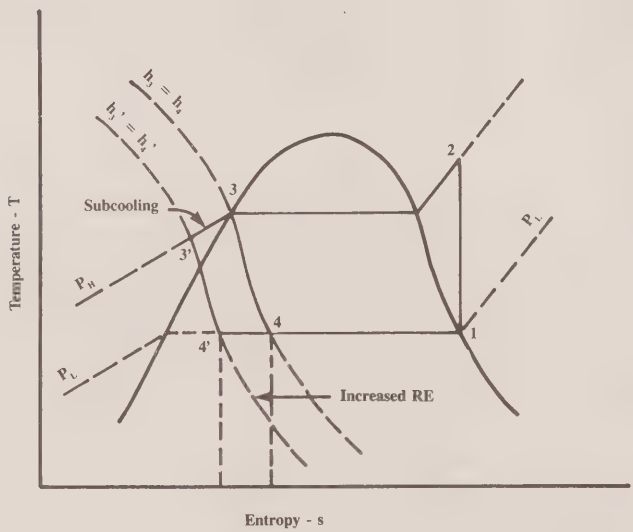
Desuperheating the high-pressure refrigerant (hot gas) leaving the compressor will reduce the required condenser capacity, and provide a high-grade heat source for other process use. A typical application would be the preheating of boiler make-up or process water. The total amount of heat available as superheat can be difficult to predict, as the superheat fluctuates with changes in load conditions. If a use can be found for low-grade heat, the total condensing load can be reclaimed. This can result in substantial energy savings.

Flash Gas and Subcooling

Liquid subcooling occurs when a liquid refrigerant is cooled at constant pressure to below the condensation temperature (Figure 17). When subcooling occurs by a heat transfer method *external* to the refrigeration cycle, the refrigerating effect of the system is increased because the enthalpy of the subcooled liquid is less than the enthalpy of the saturated liquid. Subcooling of the liquid upstream of the throttling device also reduces *flashing* in the liquid piping. The work input is reduced, and the refrigeration effect is increased because $(h_1 - h_4)$ is less than $(h_1 - h_4')$.

Subcooling refrigerant R-22 by 13°C increases the refrigeration effect by about 11 per cent. If subcooling is obtained from *outside* the cycle, each degree increase in subcooling will improve system capacity by approximately one per cent. Subcooling from *within* the cycle may not be as effective because of offsetting effects in other parts of the cycle.

Subcooling capacity can be increased by providing additional cooling circuits in the condenser or by immersing the liquid receiver in a cooling tower sump. Most systems provide 5 to 7°C subcooling at the condenser to improve system efficiency.



Effect Of Subcooling
Figure 17

Hot Gas Bypass

Hot gas bypass is a method of placing an artificial heat load on the refrigeration system to produce stable suction pressures and temperatures, when the refrigeration load is very low. The heat load is produced by bypassing hot gas from the compressor discharge to the evaporator inlet or the compressor suction. While permitting stable compressor operation at low load, hot gas bypass wastes energy. Bypass is required to maintain evaporator temperature above freezing, and prevent frosting of the coil, freezing of the chilled water, and compressor cycling.

The total refrigeration load on a compressor with hot gas bypass will be equal to the actual (low) load plus the amount of hot gas bypass. Typically, the hot gas bypass on a reciprocating machine is 25 per cent of the nominal refrigeration capacity for a 4 cylinder unit, 33 per cent for a 6 cylinder unit and 37.5 per cent for an 8 cylinder unit. For centrifugal equipment, the bypass varies with the load and impeller characteristics.

Evaporator Frosting

When a refrigeration system operates with the evaporator temperature close to 0°C, or slightly less, frosting of the evaporator coil is inevitable. Examples of this would be the frosting of heat pump evaporator coils during winter operation, or freezer evaporators. Ice buildup on the coils lowers the heat transfer rate, effectively reducing the refrigeration effect. The suction temperature will fall as the heat transfer rate falls, further increasing the rate of ice buildup. For systems operating under these conditions defrosting accessories are available from the equipment manufacturer.

Defrost is performed by reversing the refrigerant flow, so that the system operates in an air-conditioning mode, using the evaporator as the condenser to reject heat through the frosted coils. In a heat pump system used for heating, a back-up heating system is required to prevent chilling the space during the defrost mode. Defrosting is a major consumer of energy. It is important that the controls optimize the defrost cycle to avoid unnecessary defrosting while preventing unwanted ice buildup.

Heat Pump Cycle

The heat pump is a separate class of compression refrigeration equipment whose main purpose is to transfer heat from a low temperature heat source to a higher temperature heat sink for heating, rather than for cooling. The coefficient of performance in the heating configuration is:

$$\text{COP(Heat Pump)} = \frac{\text{Refrigeration effect plus work input}}{\text{Net work input}}$$

$$= \frac{T_H}{(T_H - T_L)}$$

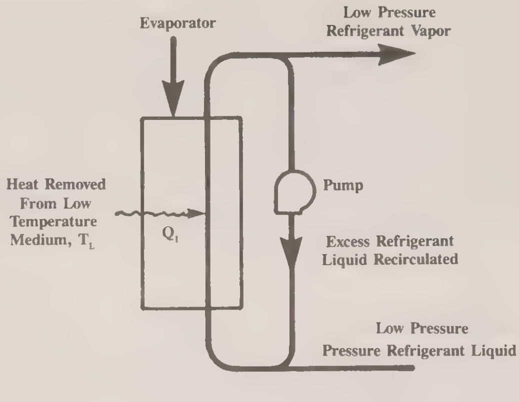
In a heat pump system where both heating and cooling are required, a special four-way valve is used to reverse the functions of the evaporator and condenser. In this manner, the coil or exchanger is used to supply heating or cooling as required. Alternatively, the piping or ductwork system *external to the heat pump* can be provided with valves or dampers to reverse the primary air or fluid flows, without the reversing valve. The heat pump cycle is identical to a standard refrigeration cycle on a T-s diagram (Figure 13).

Absorption Cycle

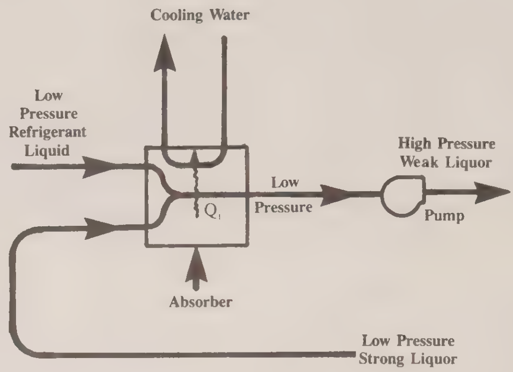
The *absorption* refrigeration cycle is similar to the vapor compression cycle, however, instead of using a compressor, high pressures are obtained by applying heat to a refrigerant solution.

The system operates on the principle that variations in refrigerant solubility can be obtained by changing solution temperatures and pressures. Absorption systems in industry often use ammonia as the refrigerant in a water solvent, whereas in commercial and institutional applications water is used as the refrigerant in a lithium bromide solvent.

The basic components of an absorption system are the vapor absorber, solution transfer pumps, and a vapor regenerator (solvent concentrator) in addition to the evaporator and condenser.



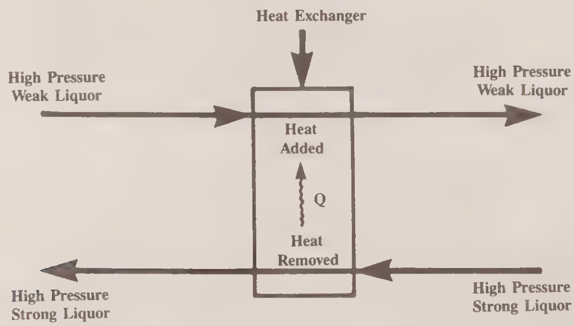
Evaporation
Figure 18



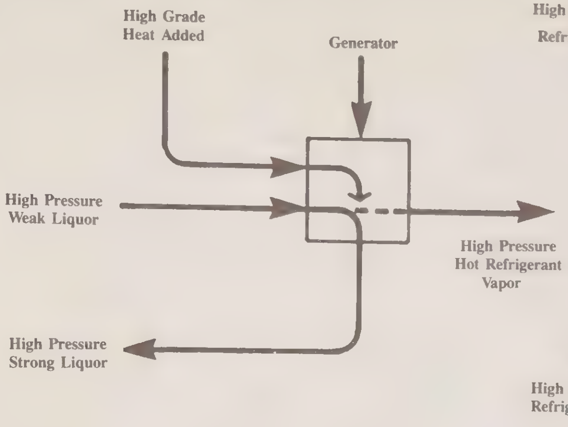
Absorption
Figure 19

The steps in an absorption refrigeration cycle are:

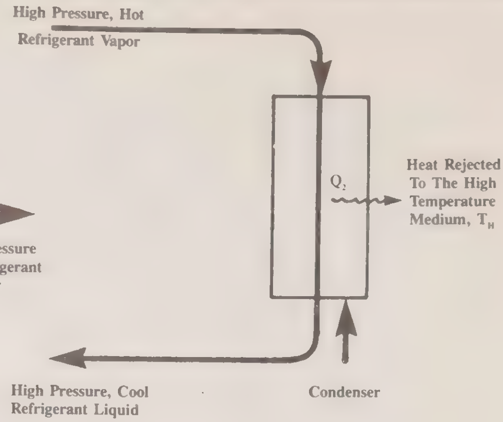
1. Liquid refrigerant is vaporized in the evaporator, absorbing heat from the medium to be cooled (Figure 18).
2. The suction effect necessary to draw the vapor through the system is accomplished by bringing the refrigerant into contact with a solvent (Figure 19). The solvent's *affinity* for the refrigerant causes the refrigerant to be absorbed by the solution, reducing the pressure of the refrigerant vapor. The absorption process *releases* some heat which must be removed from this portion of the cycle. The solution of refrigerant and solvent (weak liquor) is pumped from the absorber at low pressure, to the generator at a high pressure.
3. Heat is added to the weak liquor to drive the refrigerant out of solution. A heat exchanger is located between the absorber and generator (Figure 20). Heat is removed from the strong liquor (solution with high solvent and low refrigerant concentrations) leaving the generator, and is added to the weak liquor entering the generator, reducing the cycle heat input.
4. Further heat added to the weak liquor in the generator drives the refrigerant out of solution, providing a high pressure refrigerant vapor (Figure 21). The hot solvent, still containing some refrigerant (strong liquor), returns to the absorber through the heat exchanger where the solvent cycle repeats.
5. Vapor at high-pressure and temperature flows to the condenser (Figure 22) where heat is rejected through a coil or heat exchanger during the condensation process.
6. The pressure of the liquid refrigerant is reduced by passing through a throttling device before returning to the evaporator section. The complete cycle is shown in Figure 23,



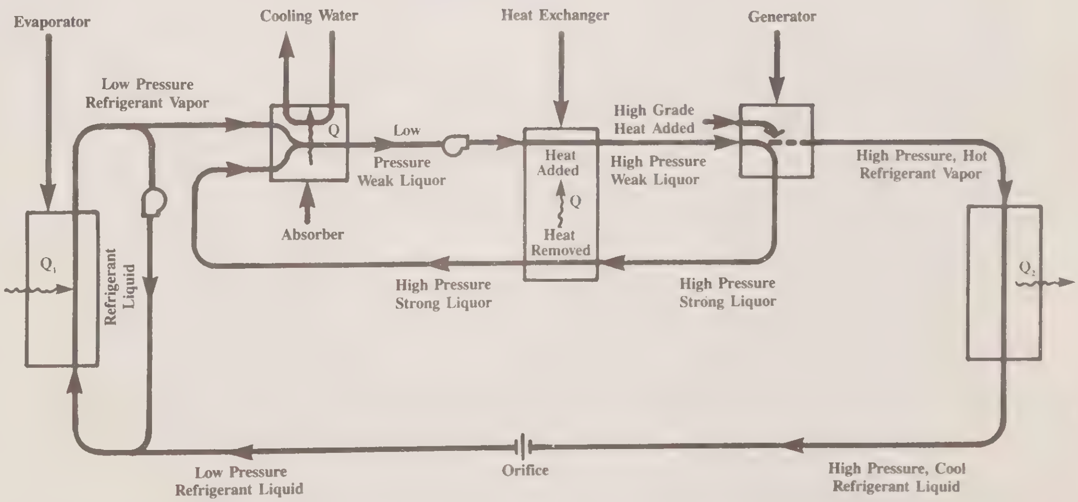
Heat Added To Weak Liquor
Figure 20



Refrigerant Driven Out Of Solution
Figure 21



Condensing
Figure 22



Absorption Refrigeration Cycle
Figure 23

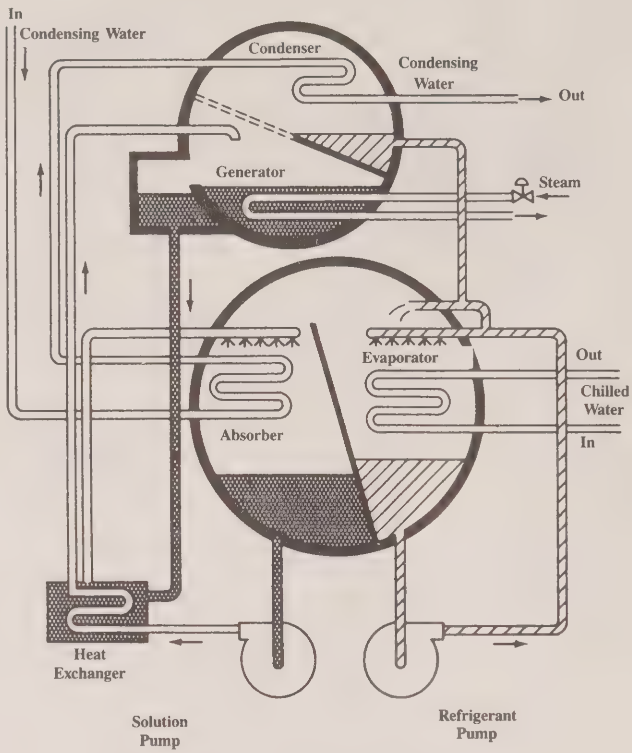


Diagram of a Two-Shell Lithium Bromide Cycle Water Chiller
Figure 24

Courtesy of ASHRAE

The generator may be equipped with a *rectifier* for selective distillation of refrigerant from the solution. This feature is common in large ammonia systems.

Performance of an absorption chiller is measured by the COP, the ratio of actual cooling or heating effect, to the energy used to obtain that effect. The best ratios are less than one for cooling and 1.2 to 1.4 for a heat pump application. Compared to compression cycles this is low, but if high-temperature waste heat can be utilized to regenerate the refrigerant, refrigeration can be obtained at reasonable costs.

System performance is affected by:

- Heat source temperature.
- Temperature of medium being cooled.
- Temperature of the heat sink.

The flow diagram of a two-shell lithium bromide chiller is shown in Figure 24. Figure 25 shows an alternative configuration of an absorption machine using only a single shell. Actual installations vary considerably in layout, number of components and accessories, application and refrigerant type.

Special Refrigeration Systems

Steam jet refrigeration systems use steam ejectors to reduce the pressure in a tank containing the return water from a chilled water system. *Flashing* a portion of the water in the tank reduces the liquid temperature. The chilled water is then used directly or passed through an exchanger to cool another heat transfer fluid.

Well water, or any other clean water below 15°C, can be used for cooling or precooling ventilation air, or a process.

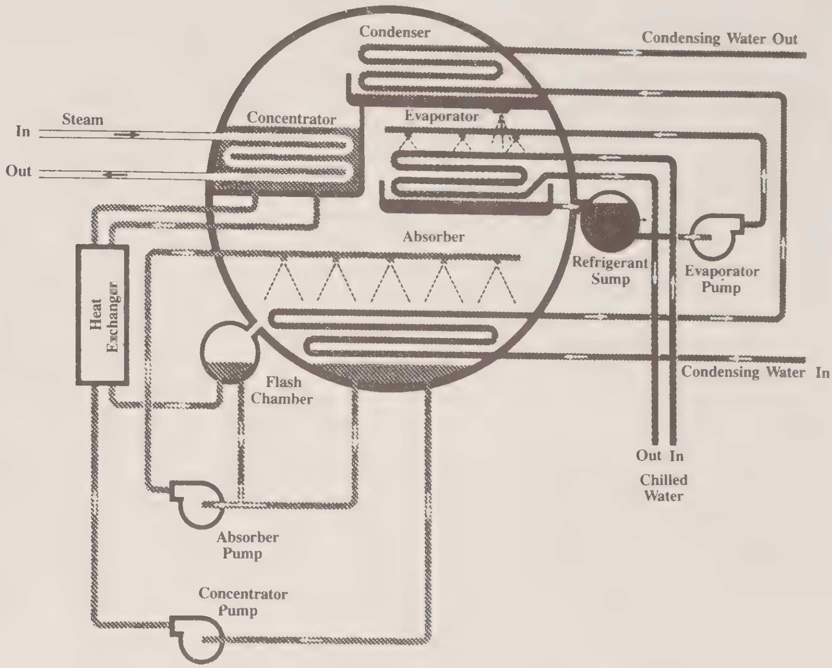


Diagram of a One Shell Absorption Machine
Figure 25

Refrigerants

Desirable Characteristics

Refrigerants for Industrial, Commercial and Institutional refrigeration and heat pump systems are selected to provide the best *refrigeration effect* at a reasonable cost. The following characteristics are desirable.

- Nonflammable to reduce the fire hazard.
- Nontoxic to reduce potential health hazards.
- Large heat of vaporization to minimize equipment size and refrigerant quantity.
- Low specific volume in the vapor phase to minimize compressor size. This aspect is critical for reciprocating and screw type compressors.
- Low liquid phase specific heat to minimize the heat transfer required when subcooling the liquid below the condensing temperature.
- Low saturation pressure required at desired condensing temperatures to eliminate requirement for heavy duty or high pressure equipment.
- Low pressure portion of the cycle should be above atmospheric pressure to prevent inward leakage of air and water vapor into the refrigerant piping.
- High heat transfer coefficients.

The American National Standards Institute (ANSI) and The American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc. (ASHRAE) have prepared lists of standard designations for refrigerants. Each refrigerant is assigned a "refrigerant number" characteristic of its chemical structure. Table 3 lists some of these designations.

Physical properties of various common refrigerants are listed in Table 4.

The relative safety and hazard level of various refrigerants have been compiled and classified under ANSI Code B9.1 - 1971 and by Underwriter's Laboratories. Table 5 provides a listing of these properties for various refrigerants.

Common Refrigerants - Vapor Compression Cycles

Fluorinated halocarbons are the most common refrigerants. They are nontoxic, nonflammable, noncombustible, and noncorrosive, provided that the refrigerants are dry. In concentrations of less than 20 per cent by volume in air, they are odorless. Above this concentration, mild ether-like odors result. Typical of this class are refrigerants R-11, R-12 and R-22, commercially known as Freon 11, 12 and 22, or Genetron 11, 12 and 22, respectively.

Ammonia, refrigerant R-717, one of the earliest refrigerants, is now limited to industrial applications because of its high toxicity. High cycle efficiency, low specific volume, high latent heat and low cost led to its popularity, particularly in ice rink facilities and other applications where large temperature differences were required.

Carbon dioxide is a nontoxic, nonflammable, odorless, colorless, and inert gas. Because of high operating pressures and high horsepower requirements its use as a refrigerant is limited to specific industrial applications.

Common Refrigerants - Absorption Cycle

Ammonia is a refrigerant used with *water* as the absorbent (solvent). Use of ammonia is declining with the introduction of refrigerants that have low toxicity and operate at lower system pressures.

Water, is the most common refrigerant, and is used in combination with *lithium bromide* as the absorbent.

Brines and Secondary Coolants

Secondary refrigerants, brines and heat transfer fluids find common use in refrigeration applications. These liquids are cooled or heated by the primary refrigerant and transfer heat energy without a change of state. Their use is common where:

- Large refrigerant quantities would otherwise be required.
- Toxicity or flammability of the refrigerant is a concern.
- Central refrigeration is used to produce cooling for a number of remote locations.

Many examples exist where brines and secondary coolants are used.

- Chilled water or glycol-water solutions for air-conditioning and process cooling.
- Calcium chloride or sodium chloride in solution with water for ice manufacture in skating rink applications.
- Propylene glycol and water solutions for use in food and potable water refrigeration systems.
- Hydrocarbon refrigerants in the liquid phase for extremely low temperature applications.

Selection of the brine type and concentration is made on the basis of freezing point, crystallization temperature, specific volume, viscosity, specific heat and boiling point. Toxicity, flammability and corrosion characteristics are secondary factors, but must be considered in the overall analysis.

Energy Audit Methods

Energy Management Opportunities exist in refrigeration and heat pump systems used in Industrial, Commercial and Institutional facilities. Many of these opportunities are recognizable during a *walk through* audit of the facility. This audit is usually more meaningful if a "fresh pair of eyes' generally familiar with energy management is involved. Typical energy saving items noted during a walk through audit are refrigeration systems operating when not required, leaking chilled water piping, low chilled water temperatures and high condenser water temperatures. Alert management and operating staff, and good maintenance procedures can reduce energy usage and save money.

Not all items noted in a the walk through audit are as easy to analyze as those described. For example, it may be observed that the chilled water supply temperature is 4.5°C. The immediate reaction is that the water temperature could be increased to improve the refrigeration plant operating efficiency. This leads to the following questions:

- How much can the temperature be increased?
- Will cooling capacity be reduced?
- Is dehumidification critical?
- Can decentralized cooling systems be provided to solve local cooling problems?
- Will the energy cost savings pay for the changes with additional return in the future?

This example requires a *diagnostic audit* to mathematically establish potential reductions in energy use obtained from changes of evaporator temperature, and to calculate potential dollar savings. Simple payback calculations can be performed to establish the financial viability of the opportunity once the estimated cost of the changes is established.

The implementation of Energy Management Opportunities can be divided into three categories.

- *Housekeeping* refers to an energy management action that is *repeated on a regular basis and never less than once a year*. An example of this could be the cleaning of heat exchangers in a refrigeration system to maintain high heat transfer rates.
- *Low cost* refers to an energy management action *that is done once and for which the cost is not considered great*. Examples of low cost items would be the installation of control valves to shut off the water supply to an evaporative cooler when sufficient cooling can be obtained by using a dry coil, or the reclaim of condenser heat using auxiliary condensers.
- *Retrofit* refers to an energy management action *that is done once and for which the cost is significant*. Examples would be the installation of a heat pump system to reclaim waste heat, and the upgrading of an absorption system to a centrifugal system because low cost waste heat is no longer available.

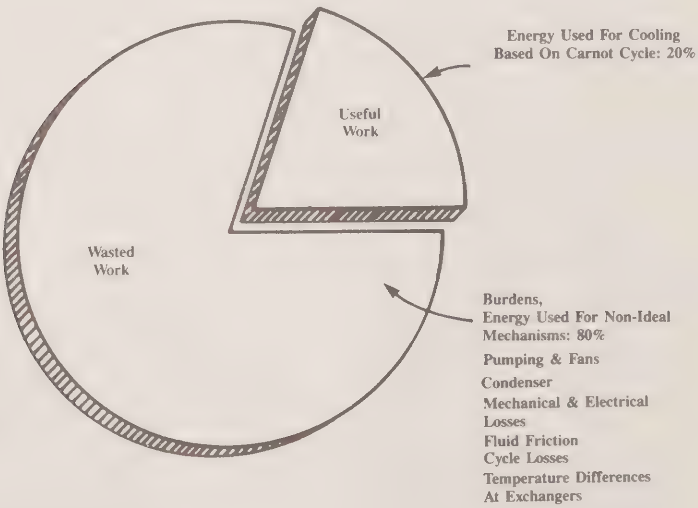
It must be noted that the division between low cost and retrofit is normally a function of both the size and the type of the organization as well as its cash flow position.

Fundamentals Summary

Fundamental principles of refrigeration system operation have been discussed to focus attention on the various areas where Energy Management Opportunities exist.

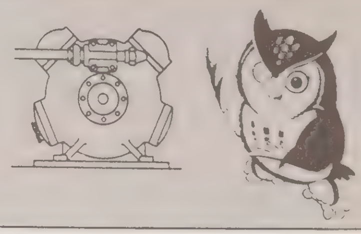
If the *effectiveness* of a system is defined as the ratio of the actual COP to the Reverse Carnot cycle COP, then the typical system effectiveness is in the range of twenty to twenty-five per cent. By applying energy management techniques, the effectiveness of a system can be increased and energy use reduced.

The pie chart (Figure 26) shows the overall effect of system inefficiencies and physical restraints on system effectiveness. Large gains can be made by improving specific components. For example, an improvement in mechanical and electrical efficiency can reduce the energy input to the system for the same refrigerating capacity.



Energy Balance For Reciprocating Compressor Refrigeration Systems
Figure 26

EQUIPMENT SYSTEMS



The following major components are required in vapor compression refrigeration systems.

- Refrigerant compressors.
- Evaporators.
- Throttling devices.
- Condensers.
- Heat rejection equipment.

Refrigerant Compressors

Displacement and dynamic compressors are commonly used for refrigeration.

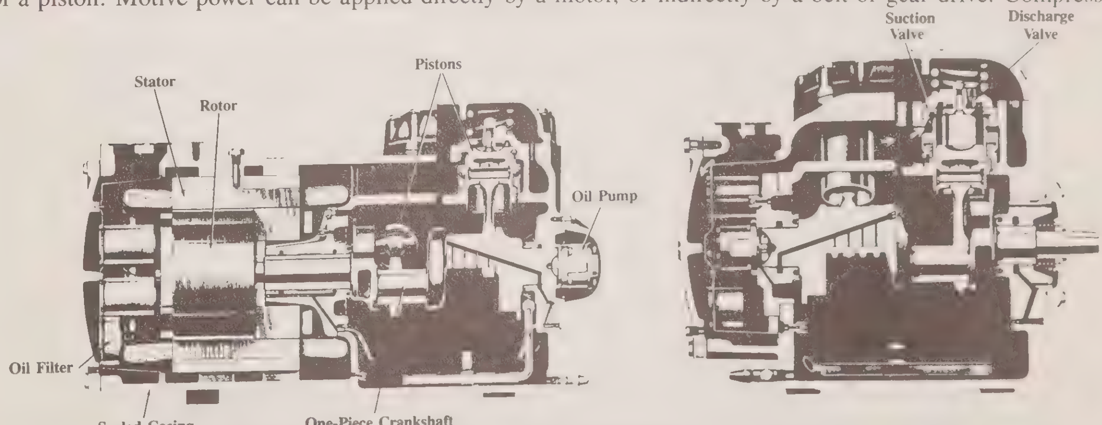
- *Displacement* machines increase the refrigerant pressure by reducing the volume of the compression chamber. This is done by applying shaft work to the mechanism. This category includes reciprocating, rotary (vaned), and screw (helical rotary) compressors.
- *Dynamic* machines increase refrigerant pressure through a continuous exchange of angular momentum between a rotating mechanical element and the fluid being compressed. Centrifugal and turbo compressors are in this category.

Displacement and dynamic refrigeration compressors are classified as hermetic, semi-hermetic or open.

- *Hermetic* compressors, including motor and drive, are sealed in a welded casing to contain the refrigerant and lubricating oil. They are available only in small sizes, are reasonably dependable and low in cost, but impractical to service.
- *Semi-hermetic* compressors are similar to the hermetic type, but the motor and compressor are in a fabricated enclosure with bolted sections or access panels to facilitate servicing.
- *Open* compressors are characterized by an external drive shaft that extends through a seal in the compressor housing.

Displacement Machines

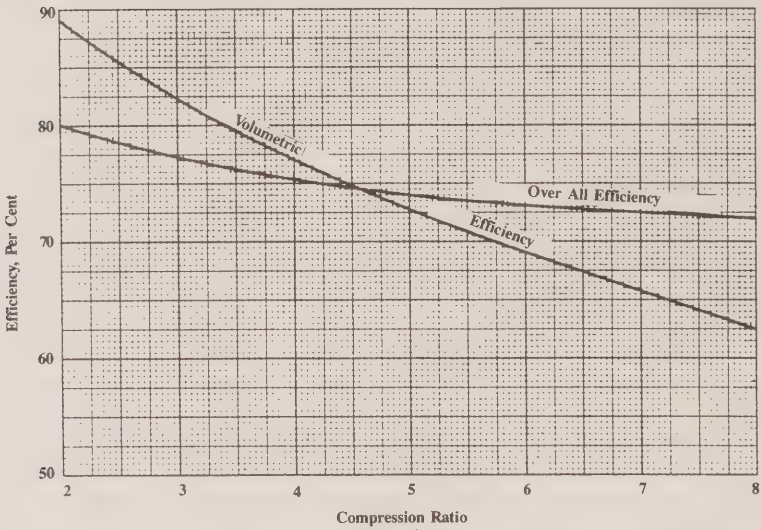
Most *reciprocating* compressors (Figure 27) are single acting, compressing the gas only on the forward stroke of a piston. Motive power can be applied directly by a motor, or indirectly by a belt or gear drive. Compressor



Hermetic Compressor — Open Type Compressor

Typical Reciprocating Compressors
Figure 27

efficiency is affected by the cylinder clearance volume, compression ratio, amount of suction superheat, valve pressure drops and the refrigerant characteristics. Figure 28 shows the variation in overall and volumetric efficiency in reciprocating compressors using refrigerants R-12 and R-22 at varying compression ratios. When the compressor discharge temperature is low, as with R-12, the compressor is usually air-cooled to prevent compressor overheating. Water-cooling is common where high compressor discharge temperatures are encountered.

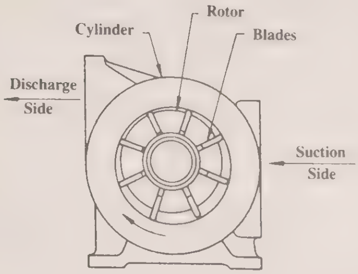


Typical Efficiencies For Reciprocating Compressors Using R-12 and R-22
Figure 28

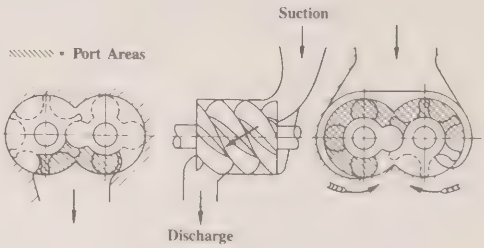
In a *rotary* compressor the rotor turns in a cavity. The gas enters through a port, the rotor seals the port and the action of the rotor compresses the gas as it is moved to the discharge port. This basic principle applies to both screw and vane compressors.

A *vane* compressor consists of a bladed, eccentric rotor in a cavity (Figure 29). As the rotor turns, the blades extend and retract, sealing off the cavity into segments of varying size. The gas enters the intake port where the segments are large, is compressed as the cavity is reduced, and discharged where the segments are small. Maximum compression ratios achieved are in the order of 7:1. Small systems and some ammonia systems use compressors of this type. In multistage systems where each stage has a low compression ratio, vane compressors are used as *boosters*.

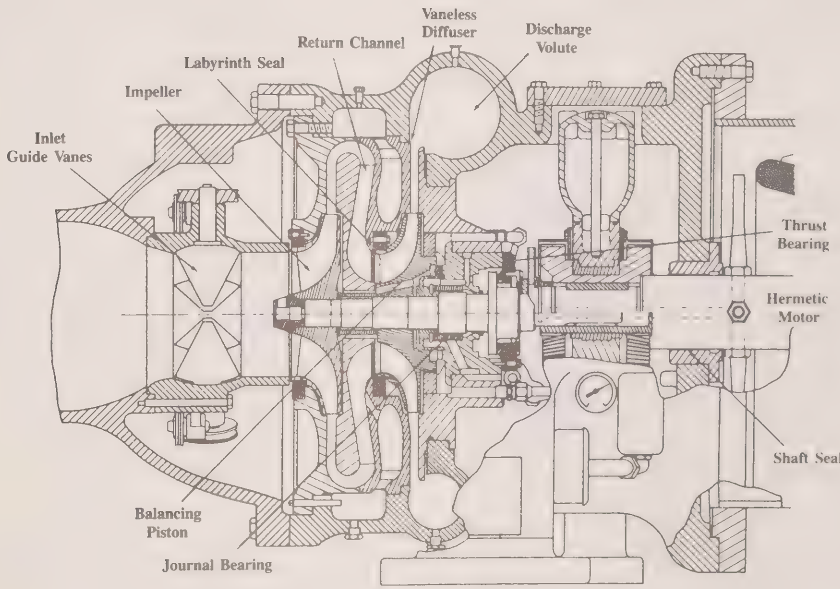
A *screw (or helical rotary)* compressor (Figure 30) consists of mating helically grooved rotors. The male rotor (with lobes) intermeshes with the female rotor (with gullies) in a stationary casing with the inlet port at one end and the outlet port at the other. As with the vane compressor, the rotating elements open a void to the suction inlet, take in a volume of gas and then seal the port. Further rotation reduces the volume between the rotors and compresses the gas. The gas is discharged at the low-volume, high-pressure end of the compressor through the outlet port. Generally, the male rotor is direct driven. The female rotor rotates along with the male, either through a gear drive, as in a dry type machine, or through direct rotor contact, as in oil-flooded equipment.



Interior View Of A Large Rotary Compressor
Figure 29



Helical Rotary Twin Screw Compressor
Figure 30



Centrifugal Refrigeration Compressor
Figure 31

Courtesy of ASHRAE

Dynamic Machines

A *centrifugal* compressor (Figure 31) operates on the same principle as a centrifugal pump, but pumps a refrigerant gas instead of a liquid. A rotating impeller imparts velocity to the gas, "flinging" it outwards. The casing slows the gas flow, converting a portion of the kinetic energy (velocity pressure) into static pressure. This type of compressor is practical for large capacity units with a low pressure ratio. Typical capacities range from 350 kW to over 10 000 kW of cooling.

A comparison between the four common types of refrigeration compressors is provided in Table 6.

COMPARISON BETWEEN FOUR COMMON TYPES OF REFRIGERATION COMPRESSORS
TABLE 6

Advantages	Disadvantages
Rotary and Vane	
• Good efficiency as booster:equal to screw and better than piston type • Handles low pressure conditions • Mechanically reliable	• Discharge pressure limitation • Overall pressure ratio limited to about 7:1 • Poor part load power characteristics
Reciprocating Piston	
• Basic industry work horse • Full range of sizes & capacities • Efficient part load operation • Relatively inexpensive • Requires minimum amount of support infra-structure	• Volumetric efficiency drops at high overall pressure ratios • Requires frequent maintenance • Not tolerant of liquid • Water cooling necessary for ammonia systems
Rotary Screw	
• Good efficiency at full load • Large capacity units available • Low maintenance costs • Reliable • Tolerant to liquid • Liquid injection cooling option • Infinitely variable capacity control • High operating flexibility	• Poor power performance at low part load conditions • Small sizes expensive • Repairs expensive in remote locations
Centrifugal	
• Efficient at full load • Large capacity units require small space	• Very high speed precision equipment • Useable only with freon type refrigerants • Inefficient at part load • Severe operating restrictions

Evaporators

Evaporators commonly used in process cooling and air-conditioning systems fall into two categories.

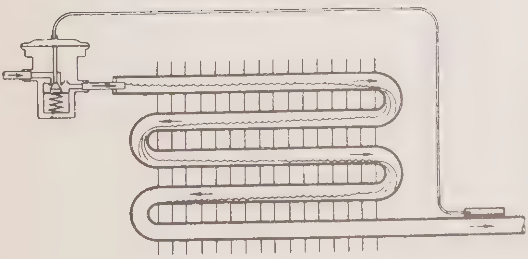
- Direct expansion (DX) coils - primarily used for cooling air or other gas streams.
- Liquid coolers (shell and tube water chillers) - primarily used for producing chilled water, glycol, or brine for process cooling or air-conditioning.

Coils

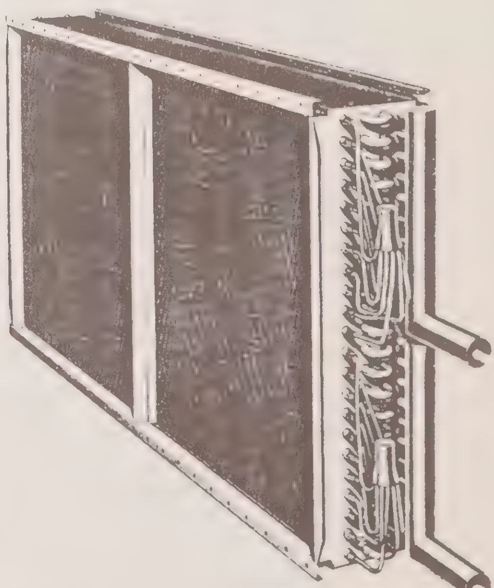
Direct expansion (DX) coils (Figure 32) consist of a series of tubes through which refrigerant flows. The tubes are arranged into a number of parallel circuits fed from a single expansion valve. The hot refrigerant vapor is collected in the outlet (suction) gas header. The tubes are finned to increase the heat transfer rate from the medium to be cooled, generally air, to the boiling refrigerant.

DX coils are used only in positive displacement compressor systems because dynamic machine pressure-ratios are too low for proper coil operation. DX coils are subdivided into two types, flooded and dry.

- With a *flooded* coil, a float valve maintains a preset liquid level in the coil, keeping the evaporator coil nearly full of liquid refrigerant. This full contact of the liquid with the tube walls ensures a high rate of heat transfer. However, because of the large quantities of refrigerant required, flooded type evaporators are often impractical.
- *Dry* coils contain little liquid refrigerant, reducing the cost of the refrigerant charge. A metering device, called a thermal expansion (TX) valve, regulates the amount of liquid that enters the coil in order to maintain a predetermined amount of superheat in the refrigerant at the coil outlet. A dry expansion coil contains mostly liquid at the inlet and only superheated vapor at the outlet, after having absorbed heat from the medium to be cooled.



Thermostatic Expansion Valve Controlling Flow Of Liquid Refrigerant Entering Evaporator



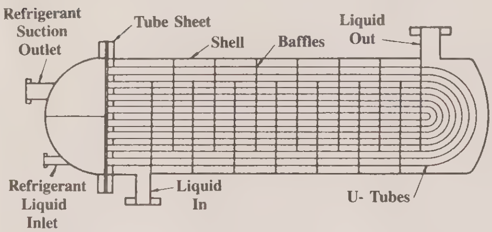
Dry Expansion Coil

Direct Expansion Coil Details
Figure 32

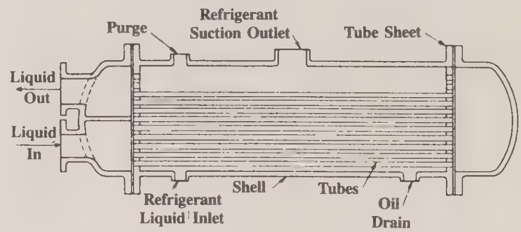
Liquid Coolers

Shell and tube heat exchangers (Figure 33) are used to cool liquids, which can be used as a secondary refrigerant, or to cool the final product directly. These exchangers are referred to as chillers or liquid coolers. Applications include:

- Chilling water for air-conditioning coils.
- Chilling milk after the pasteurization process.
- Chilling water in a drinking fountain.
- Process cooling.



Direct (U Tube-Type) **Expansion Liquid Cooler**



Ammonia Shell-and-Tube Liquid Cooler

Types of Liquid Coolers

Type of Cooler	Usual Refrigerant Feed Device	Usual Range of Capacity tons (kW)	Commonly Used with Refrigerant Nos.
Flooded shell-and-bare-tube	Low-pressure float	50-500 (175-1750)	717 (Ammonia)
Flooded shell-and-finned-tube	Low-pressure float; High-pressure float, fixed orifice(s), wier(s)	25-2000 (175-35 000)	11, 12, 22, 113, 114, 500, 502
Spray-type shell-and-tube	Low-pressure float; High-Pressure float	50-10 000 (350-1750)	11, 12, 13 B1, 22, 113, 114
Direct-expansion shell-and-tube	Thermal expansion valve	5-350 (17.5-1250)	12, 22, 500, 502, 717
Flooded Baudelot cooler	Low-pressure float	10-100 (35-350)	717
Direct-expansion Baudelot cooler	Thermal expansion valve	5-25 (17.5-85)	12, 22, 717
Flooded double-pipe cooler	Low-pressure float	10-25 (35-85)	717
Direct-expansion double-pipe cooler	Thermal expansion valve	5-25 (17.5-85)	12, 22, 717
Shell-and-coil cooler	Thermal expansion valve	2-10 (7-35)	12, 22, 717
Flooded tank-and-agitator	Low-pressure float	50-200 (175-700)	717

Liquid Coolers
Figure 33

Courtesy of ASHRAE

Liquid coolers are subdivided into two categories.

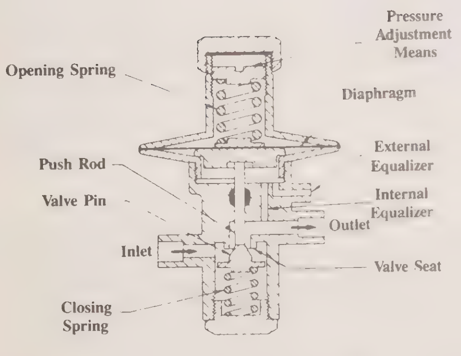
- In a *dry cooler*, the liquid refrigerant is contained within tubes, and water or brine circulates through the shell of the cooler (evaporator).
- In a *flooded cooler* the water or brine circulates through the tubes. Usually, the tubes are finned to increase the heat transfer rate and reduce the evaporator size.

Dry type coolers are used in air-conditioning and refrigeration systems under 615 kW cooling, because the refrigerant charge is smaller, and the risk of equipment damage from freezing is less. Flooded coolers are more efficient on larger systems.

Throttling Devices

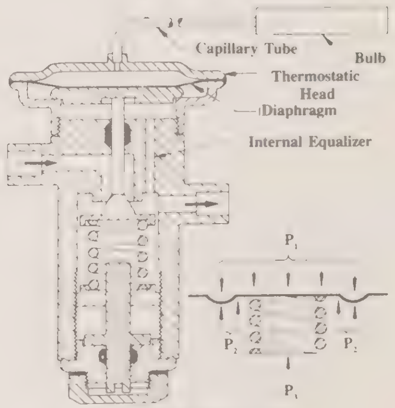
A throttling device in a vapor compression system maintains the pressure differential between the high pressure (condenser) side and the low pressure (evaporator) side. The device regulates the flow of liquid refrigerant to the evaporator to match the equipment and load characteristics.

- A *constant pressure expansion valve* (Figure 34) maintains a constant load on the compressor, regardless of the load on the evaporator, by metering the liquid refrigerant flow into the evaporator, based on suction pressure.
- The *thermostatic expansion valve* (Figure 35), the most widely used control device, automatically meters the flow of liquid refrigerant to the evaporator at a rate that matches system capacity to actual load. Connection of several DX evaporators in parallel using only one compressor is possible when each evaporator is provided with individual control. The valve senses the pressure and the temperature of the refrigerant leaving the evaporator and regulates the liquid refrigerant flow into the coil. Temperature is sensed at the coil outlet and pressure at the coil inlet.



Valve is used with either internal or external equalizer, but not with both.

Constant Pressure Expansion Valve
Figure 34

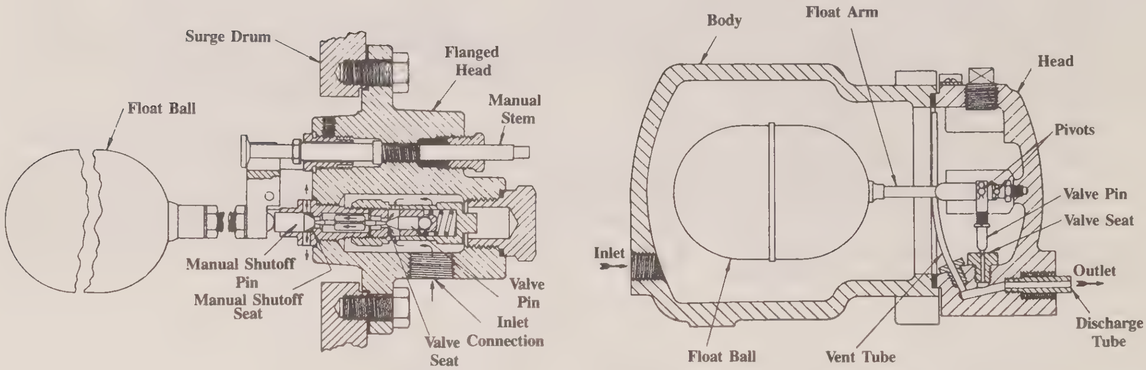


P_1 - Thermostatic Element's Vapor Pressure
P_2 - Evaporator Pressure
P_3 - Pressure Equivalent Of The Superheat Spring Force

Thermostatic Expansion Valve
Figure 35

Courtesy of ASHRAE

• *Float valves* (Figure 36) are used for metering refrigerant flow to a flooded type liquid cooler. A low-side float valve is located on the low pressure side of the throttling device. A high-side float valve is located on the high pressure side of the throttling device. Liquid refrigerant is admitted by the float valve to the evaporator shell, at the same rate at which it is removed by the compressor. In some systems, the float operates an electrical switch that controls a solenoid valve. The solenoid valve periodically admits liquid refrigerant to the evaporator, allowing the liquid level to fluctuate within preset limits.



Low-Side Float Valve

High-Side Float Valve

Figure 36

Courtesy of ASHRAE

Condensers

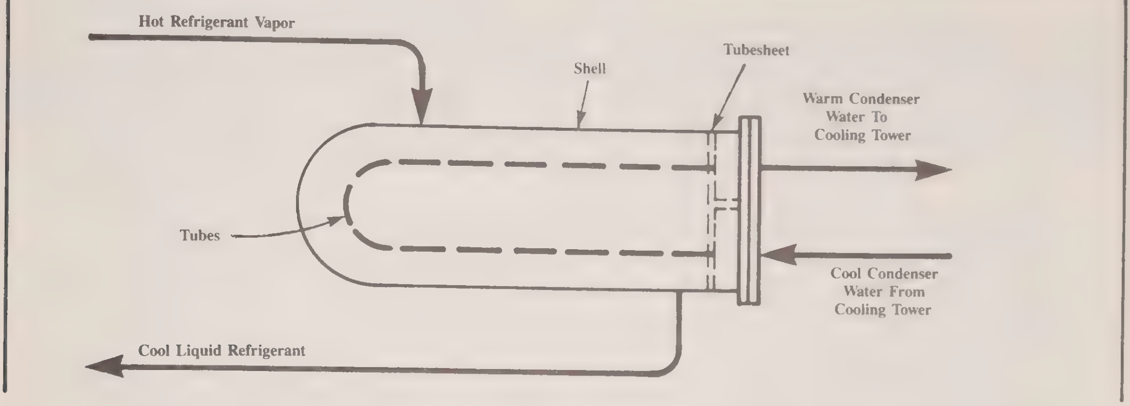
Condensers are generally shell and tube type heat exchangers with refrigerant flow through the shell and coolant flow through the tubes. The lower portion of the shell acts as a liquid receiver. Where the lower tube sections are submerged in refrigerant, the exchanger also acts as a subcooler to minimize flash gas in the throttling process.

Other configurations are air-cooled coils, and evaporative condensers where water is sprayed over the coil. Auxiliary receivers and coolers are often used with these arrangements. Fans increase air flow over the coils and thus increase the heat transfer rate.

Water-Cooled Condensers

The following points must be examined if water-cooled condensers (Figure 37) are being considered.

- A supply of cooling water for heat rejection is required.
- Where an inexpensive cooling water supply is not available, a cooling tower may prove practical.
- Auxiliary pumps and piping for recirculation of cooling water are required.
- Water treatment is required in water recirculation systems.
- Space requirements.
- Maintenance problems.
- Freeze protection for winter operation.

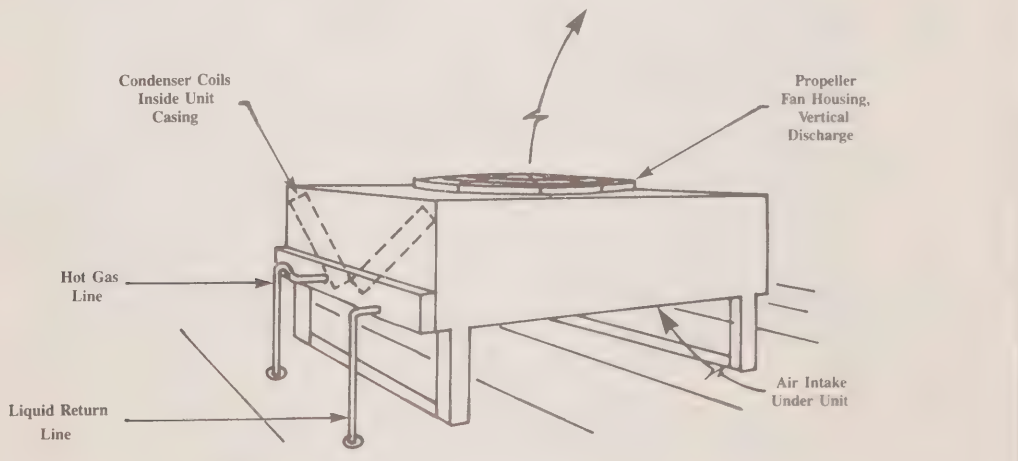


Typical Water Cooled Condenser
Figure 37

Air-Cooled Condensers

Air-cooled condensers (Figure 38) use outside air as the cooling medium. Fans draw air past the refrigerant coil and the latent heat of the refrigerant is removed as sensible heat by the air stream. Air-cooled condensers are characterized by:

- Low installation costs.
- Low maintenance requirements.
- No water requirements. (Freezing, scaling, corrosion, water piping, circulating pumps and water treatment are eliminated).
- Outdoor installation is standard.
- Higher power requirements per kW cooling than evaporative or water-cooled condensers.
- Long refrigerant lines, that require greater care in system design and installation.
- Higher refrigerant cost, because of longer piping runs.
- Operating difficulties caused by increased condensing capacity and lower loads when operating at low ambient temperatures.
- Possible noise problems because of the high air volumes required for removing condenser heat.
- Multiple units are required in large systems.

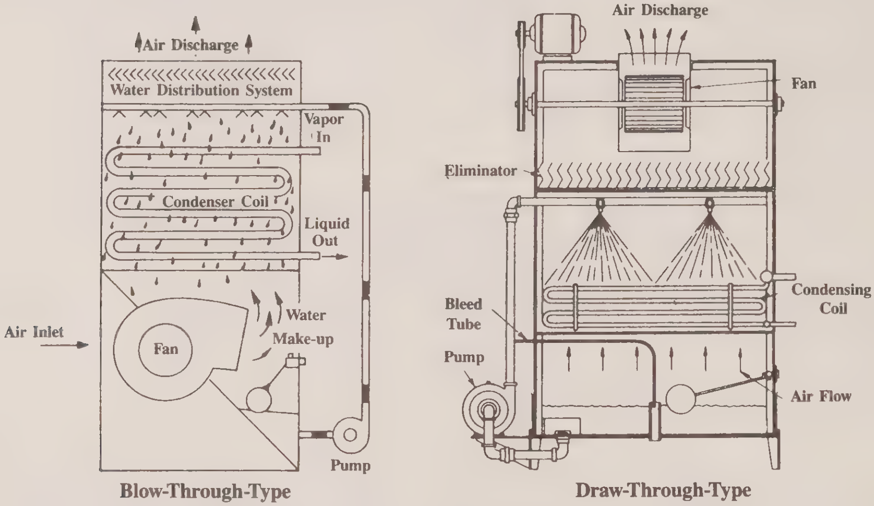


Typical Air-Cooled Condenser
Figure 38

Evaporative Condensers

In an evaporative condenser (Figure 39), a water spray is directed over a refrigerant coil. The water absorbs heat from the refrigerant vapor in the coil causing the refrigerant vapor to condense. A fan draws air over the coil to remove saturated water vapor and heat. Water is added to make up for losses owing to evaporation, drift and blowdown. In winter, evaporative condensers may be operated without water. In this case, they act as air-cooled condensers. Evaporative condensers incorporate characteristics of both air-cooled and water-cooled condensers.

- For a given capacity, less circulating water is required than for a water-cooled condenser with a cooling tower.
- Piping sizes are smaller and overall lengths are shorter.
- System pumps are smaller.
- Indoor locations are possible.
- Water treatment is required.
- Large capacity units are available.
- Space requirements are less than for air-cooled condensers, or shell and tube condensers when a cooling tower is used.



Functional View Of Evaporative Condenser
Figure 39

Courtesy of ASHRAE

Heat Rejection Equipment

Common methods of rejecting heat are:

- Air-cooling.
- Water-cooling.
- Evaporative cooling.

Air-Cooling Equipment

General discussion of air-cooled equipment is included in the "Condensers" section of this module.

Water-Cooling

A *water-cooled* system requires a clean and inexpensive source of cool water. Lake or river water may be practical only if both water treatment (required for prevention of scale buildup and corrosion protection) and filtration requirements are minimal. Municipal bylaws often limit the amount of potable water used for once-through, direct cooling, of refrigeration systems. Cooling towers and closed circuit evaporative coolers are used for rejecting heat when natural or municipal water sources are impractical.

Evaporative Cooling

In a *cooling tower* (Figure 40) water is sprayed into an airstream. The latent heat of vaporization is removed when a fraction of the water evaporates. The cooled water is then reused. A cooling tower limits the amount of make-up water to that required to offset water loss caused by *evaporation, drift, and blowdown*. Water treatment is required to prevent algae growth and scale buildup. Blowdown is required to remove minerals and water treatment chemicals which are concentrated by the evaporation of the water.

Cooling tower performance is limited by the wet bulb temperature. At wet bulb temperatures close to the dry bulb temperature the air is close to saturation and has less ability to evaporate water. For refrigeration systems, it is usually impractical to lower the condenser water temperature to within 4°C of the ambient wet bulb temperature (i.e. an *approach* temperature of less than 4°C).

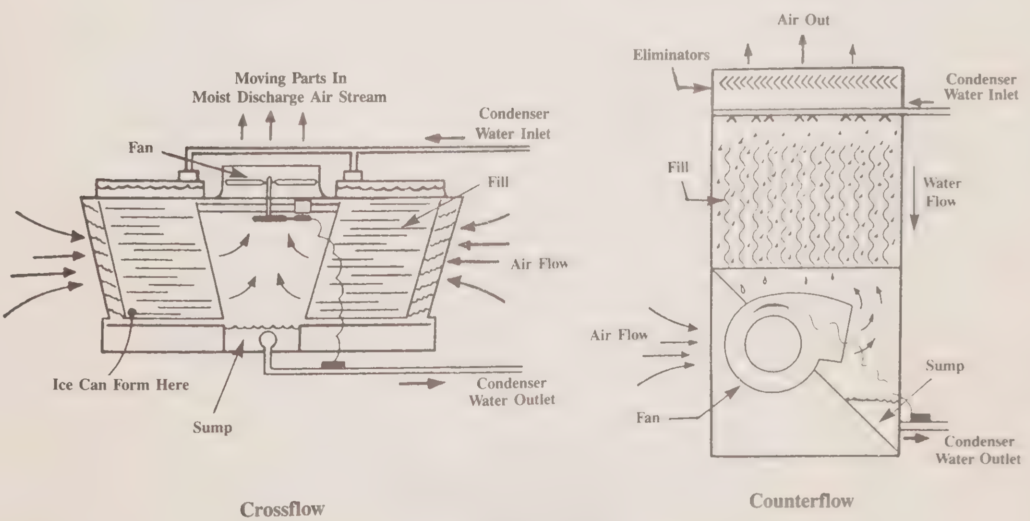
Closed circuit *evaporative coolers* use evaporation of water as the prime method of heat removal. Construction is similar to evaporative condensers, but the fluid being cooled is an intermediate heat transfer fluid, typically water or water-glycol solution. The cooled fluid is circulated to a shell and tube condenser to remove heat from the refrigeration system.

Other Cooling Systems

Other heat rejection methods are gaining acceptance.

Well water that is adequate in capacity and quality may be used for condensing purposes. Heat pump applications make it possible to extract heat from an *aquifer* for winter heating, and reject heat to the aquifer during the cooling season. In this case, the aquifer acts as the heat source, and the heat sink. Special permits for using ground water are required in most provinces.

Ground source heat pumps use the earth as the heat sink or heat source. Installation costs tend to be high owing to the extensive coil area, and excavation required. Maintenance is costly. A careful balance between heat added to, or removed from the soil must be maintained so that unit cooling capacity is not jeopardized. Using the ground coil as a condenser for rejecting heat limits the cooling performance by drying the soil in the vicinity of the coil.



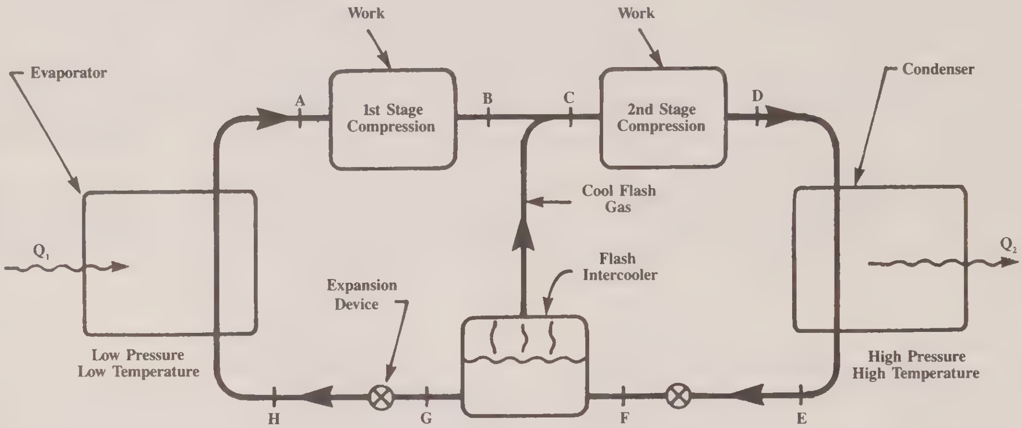
Cooling Towers

Figure 40

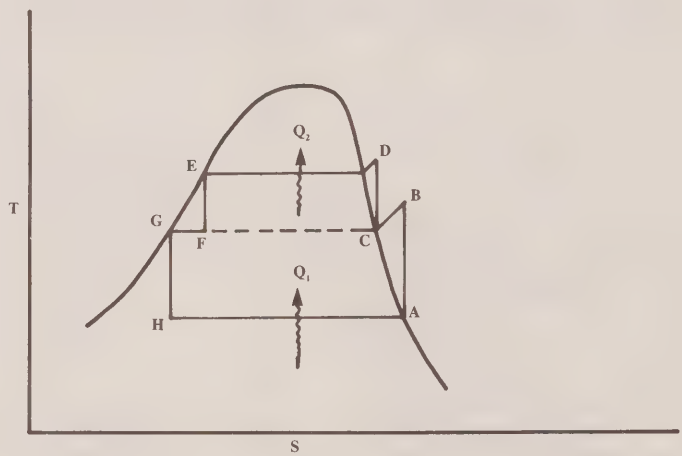
Multistage Compression Systems

A multistage system is used when large temperature and pressure differences exist between the evaporator and the condenser. Figure 41 illustrates the basic arrangement for a two-stage system where the compression work is done by either two positive displacement compressors or by two stages of a multistage centrifugal unit. The flash intercooler subcools the refrigerant liquid to the evaporator by vaporizing a portion of the refrigerant after the first throttling stage. The flash gas returns at an intermediate point in the compression process to improve the compression efficiency by cooling the superheated gas (Figure 42).

In large systems with a number of evaporators and large compression (temperature) ratios, the number of flash intercoolers and compression stages is increased to maximize system efficiency.



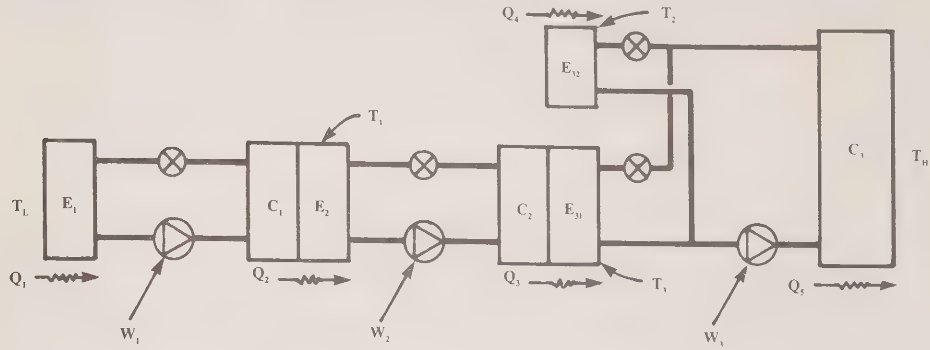
Schematic of 2-Stage Refrigeration System
Figure 41



T-s Diagram of a 2-Stage Vapor Compression Cycle. (Letters Correspond to points in Figure 41)
Figure 42

Cascaded Systems

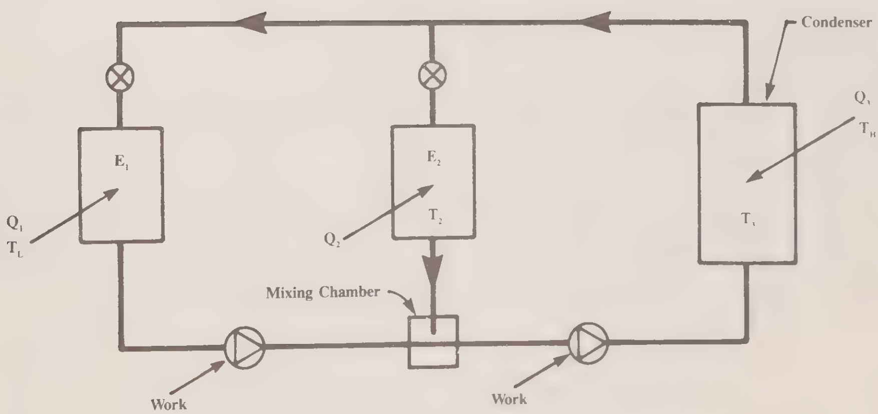
Cascaded systems are used to obtain high temperature differentials between the heat source and the heat sink. The evaporator of one system serves as the condenser to a lower temperature system. Several refrigeration systems are 'piggybacked' onto each other (Figure 43). Multiple evaporators can be used in any one stage of compression. Refrigerants used in each stage may be different and are selected for optimum performance at the given evaporator and condenser temperatures. An alternative arrangement uses a common condenser with a booster circuit to obtain two separate evaporator temperatures (Figure 44).



(W = Work, Q = Heat Transfer, E = Evaporator
C = Condenser, T_1 Above T_L, T_2 and T_3 Above T_1,
T_H = Final Condensing Temperature)

Three Stage Cascade System

Figure 43



Two Stage Cascade System
With Booster Circuit

Figure 44

Heat Pump Systems

A heat pump is a device used to transfer heat from a lower temperature to a higher temperature, for heating the warmer area or process. In many installations, *reversible* heat pumps are used, which heat or cool the process, or space.

A four-way reversing valve is used to reverse the refrigerant flow, to permit the use of the coils or exchangers in either the condenser or evaporator mode. With a fixed refrigerant circuit and no reversing valve, the secondary refrigerant flows can be reversed through appropriate external valve or damper arrangements.

Various heat source and heat sink arrangements are possible, depending on heating and cooling requirements.

- Air-to-air.
- Air-to-water.
- Water-to-air.
- Water-to-water.
- Earth-to-air.
- Earth-to-water.

In each case the first term refers to the heat source for heating applications, or the heat sink for cooling. The second term refers to the secondary refrigerant used for process or space heating and cooling. For example:

- An *air-to-air* heat pump (Figure 45) provides heating or cooling. In the cooling mode, heat is removed from the air in the space and discharged to the outside air. In the heating mode, heat is removed from the outside air and discharged to air in the space.
- An *air-to-water* system extracts heat from ambient or exhaust air to heat or preheat water used for space or process heating.
- A *water-to-air* system (Figure 46) provides heating and cooling of air with water as the heat sink or source.
- A *water-to-water* system extracts heat from a water source while simultaneously rejecting heat to a water heat sink, to either heat or cool a space or process.
- *Earth-to-air* and *earth to water* systems have limited use. Practical application is limited to space heating where the total heating or cooling effect is small, and the ground coil size is equally small.

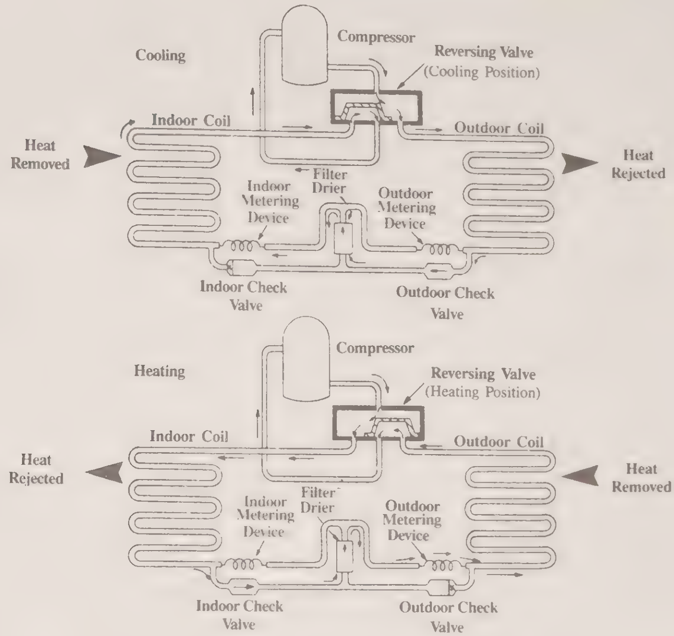
The COP for heat pump systems varies from 2 to 3 for small air-to-air space heating systems, to 5 or 6 for large systems that operate across small temperature differences.

Most heat pump systems are provided with a backup heat source to offset reductions in heat output as the evaporator (heat source, outdoor coil) temperature falls. This is particularly true in air-to-air, space-heating systems where heat output decreases as the outdoor temperature lowers.

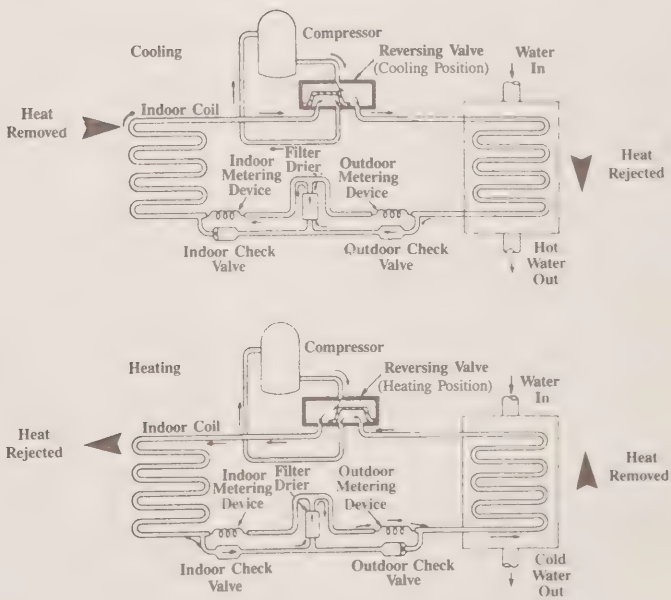
Effects of Maintenance on System Efficiency

Owners of refrigeration and heat pump equipment should follow the manufacturer's service and maintenance recommendations to maintain maximum system efficiency over the life of the equipment. Leaking seals, poor lubrication and faulty controls will reduce system life and performance.

A simple procedure, such as regular cleaning of the evaporator and condenser, has a marked effect on performance. Table 8 shows the effect of dirty heat transfer elements on an air-cooled reciprocating compressor system. Reductions in refrigerating capacity up to 25 per cent, with simultaneous increases in power input of up to 40 per cent, can result from lack of proper cleaning. Absorption chillers face reductions in refrigerating capacity of up to 24 per cent, with power increases of 7.5 per cent, from poor maintenance (Table 8).



Typical Schematic of an Air-to-Air Heat Pump System
Figure 45



Typical Schematic of Water-to-Air Heat Pump System
Figure 46

Courtesy of ASHRAE

ENERGY MANAGEMENT OPPORTUNITIES

Energy Management Opportunities is a term that represents the ways that energy can be used wisely to reduce operating costs. A number of Energy Management Opportunities, subdivided into Housekeeping, Low Cost, and Retrofit categories are outlined in this section, with worked examples or written text to illustrate the potential energy savings. This is not a complete listing of the opportunities available for refrigeration and heat pump systems. However, it is intended to provide ideas for management, operating, and maintenance personnel to allow them to identify other opportunities that are applicable to a particular facility. Other modules in this series should be considered for Energy Management Opportunities applicable to other types of equipment and systems.

Housekeeping Opportunities

Implemented housekeeping opportunities are Energy Management actions that are done on a regular basis and never less than once a year. The following are typical Energy Management Opportunities in this category:

- Keep heat transfer surfaces of evaporators and condensers clean, through regular inspection and cleaning. Fouling of the surfaces reduces the heat transfer efficiency, requiring higher temperature differences to maintain the heat transfer rate. An increase in temperature difference reduces the COP.
- Repair suction and liquid line insulation to reduce superheating of suction gas and loss of subcooling. Refrigerant lines gain heat when they are located in spaces that are not air-conditioned, increasing the system load without producing useful cooling.
- Calibrate controls and check operation on a regular basis to ensure that the refrigeration and heat pump systems operate efficiently.
- Maintain specified refrigerant charge in refrigeration and heat pump equipment. Insufficient refrigerant reduces system performance and capacity. Redused mass flow rates of refrigerant causes excessive superheating of the refrigerant at the evaporator which reduces the efficiency of the compressor, and increases the condensing temperatures.
- Provide unrestricted air movement around condensing units and cooling towers to eliminate short circuiting of the airstreams which causes a higher condensing temperature and pressure.
- Minimize the simultaneous operation of heating and cooling systems. Strategically located thermometers will help identify this problem.

Housekeeping Worked Examples

Worked examples are used to illustrate potential energy and cost savings. The examples are considered typical of conditions found in refrigeration and heat pump systems.

1. Reduce Condensing Temperature

Over time the performance of a 175 kW refrigeration system, with an air-cooled, packaged condensing unit, deteriorated. Investigation revealed that the space where the condensing unit was located had been converted to a storage area with stacked materials. Air flow to the condenser was blocked, causing short circuiting of the cooling air stream.

On a day when the ambient temperature was 35°C, the air entering the condenser was 46.1°C. The actual refrigerating load was 120 kW. Manufacturer's data for 120 kW cooling indicates that the compressor power is 42.3 kW at 35°C, and 49.76 kW at 46.1°C. The system operates 2000 hours per year at the elevated temperature. Removal of the stored materials from the condenser vicinity would prevent short circuiting and lower the air temperature entering the condenser to the ambient temperature. Electricity cost is $0.05/kWh.

Compressor energy required at 46.1°C = 2000 x 49.76

= 99 520 kWh

Compressor energy required at 35°C = 2000 x 42.3

= 84 600 kWh

Energy Saved = 99 520 − 84 600

= 14 920 kWh

Dollar Savings = 14 920 kWhx $0.05/kWh

= $746/yr

2. Clean Evaporators and Condensers

An 880 kW centrifugal chiller with a forced draft cooling tower is used to produce chilled water for air conditioning. On a walk-through audit it was noticed that algae was growing on the wetted surfaces of the cooling tower. Water blowdown to control mineral deposits and chemical feed was performed by leaving the blowdown valve open. Chemical testing and treatment was neglected.

During a plant shutdown, the heat exchanger surfaces of the evaporator and condenser were examined and found to be fouled. A contractor was hired to clean the equipment at a cost of $1,700 for each heat exchanger and $1,400 for the cooling tower, for a total of $4,800. Electricity cost is $0.05/kWh.

Performance of the system was evaluated, before and after the cleaning, using manufacturer's data and estimated COP values.

"Dirty" refrigerant suction temperature: 1.7°C = 274.7 K

"Dirty" refrigerant condensing temperature: 46.1°C = 319.1 K

"Clean" refrigerant suction temperature: 7.2°C = 280.2 K

"Clean" refrigerant condensing temperature: 40.6°C = 313.6 K

$$\text{"Dirty" COP} = 0.25^* \times \frac{T_L}{(T_H - T_L)}$$

$$= 0.25 \times \frac{274.7}{319.1 - 274.7} = 1.55$$

$$\text{"Clean" COP} = 0.25^* \times \frac{T_L}{(T_H - T_L)}$$

$$= 0.25 \times \frac{280.2}{313.6 - 280.2} = 2.10$$

* COP actual values estimated as .25 x COP (theoretical)

$$\text{Change in COP} = \frac{(2.10 - 1.55)}{1.55} \times 100 = 35\% \text{ (improvement)}$$

Power required for 880 kW cooling:

$$\text{“Dirty”} \ \frac{880}{1.55} = 568 \text{ kW}$$

$$\text{“Clean”} \ \frac{880}{2.10} = 419 \text{ kW}$$

The system operates at full load for an estimated 900 hours per year. Savings because of cleaning are:

Savings = (568 - 419) kW x 900 h x \$0.05/kWh = \$6,705

$$\text{Simple payback} = \frac{\text{Investment}}{\text{Savings}} = \frac{\$4,800}{\$6,705} = 0.72 \text{ years (9 months)}$$

Low Cost Opportunities

Implemented low cost opportunities are Energy Management actions that are done once and for which the cost is not considered great. The following are typical Energy Management Opportunities in this category.

- Increase evaporator temperature to increase system COP.

  Reset the temperature of the chilled water, glycol solution or air as a function of the cooling required, to allow the evaporator temperature to rise at part loads. For example, the setting of the air temperature leaving the evaporator of an air-conditioning system can be based on the latent load requirement. As the latent load falls, less dehumidification is required, and the controls adjust the evaporator temperature upwards.

  Relocate the outdoor coil of an air-to-air heat pump to a clean exhaust airstream. A building's exhaust is warmer than the outside ambient air during most of the heating season.
- Reduce condensing temperature to increase system COP.

  Relocate air cooled condensers and heat pump outdoor coils to clean exhaust airstreams. Generally, the building's exhaust is cooler than the outside ambient air when cooling is required.

  Reduce condenser water temperature by resetting cooling tower temperature controls. Detailed analysis is required to determine whether increased performance of the refrigeration system will offset the increased power requirement of the cooling tower fan and make-up water costs.

  Provide an automatic water treatment system to add chemicals, and control blowdown, to match the water losses of cooling tower and evaporative condenser systems. Proper water treatment will maximize heat transfer effectiveness, and keep condensing temperatures low. Benefits include reduced quantities of make-up and blowdown water, and lower operating and maintenance costs.
- Reschedule production cycles to reduce peak electrical demand and make more efficient use of available cooling or heating energy. Rescheduling may permit shutdown of some compressors in multiunit systems while running others at maximum load and peak efficiency. Operation at higher efficiency may delay purchase of additional equipment when total load increases. Refer to Electrical, Module 3 and Thermal Storage, Module 19.
- Upgrade automatic controls in refrigeration plants to provide accurate and flexible operation. Solid state digital control can optimize equipment and system operation to meet load requirements with minimum power consumption, and/or shed load to reduce short term electrical peaks. Refer to Electrical, Module 3 for load shedding techniques.
- Replace high-maintenance, centrifugal compressors with compressors selected for high efficiency when operating at part load conditions.
- Upgrade insulation on primary and secondary refrigerant piping circuits. See Process Insulation, Module 1.
- Provide multispeed fan motors on cooling towers, evaporative coolers and air cooled condensers. Normally, equipment is selected to match the rarely attained peak design condition. Lower outdoor wet and dry bulb temperatures, and lower indoor loads, predominate. Reducing condenser air flow to match the capacity requirement reduces the fan power.

- Evaporative coolers and condensers operated in winter may provide adequate capacity when operated with dry coils. Maintenance, water and electrical costs can be reduced. Heat tracing and pan heaters can be turned off. The detrimental effect of icing on equipment and buildings is eliminated. Note that the reduced power requirements for fan and cirulating pumps in cooling towers and evaporative coolers may be offset by a COP decrease caused by higher condenser temperatures. Detailed analysis is required.
- Consider a new heat pump system instead of a new air conditioning system, if winter heating is required. The higher equipment cost will be offset by reduced heating costs during the winter season.
- Provide lockable covers on automatic controls and thermostats, to prevent unauthorized tampering or adjustment.
- Use clean process cooling water that normally goes to drain for evaporative condenser or cooling tower make-up water. While not conserving energy, this will reduce operating costs.
- Reevaluate the use of hot gas bypass when a refrigeration unit works at part-load for any significant period. It may be possible to eliminate the bypass feature and cycle or turn off the refrigeration system.

Low Cost Worked Examples

Worked examples are used to illustrate potential cost savings. The examples are considered typical of the conditions found in building refrigeration and heat pump systems. Worksheets are included to be used as a guide to estimate the potential cost savings of certain opportunities.

1. Water Treatment for Condenser Water

Maintain maximum heat transfer rates by minimizing fouling. Consider the condenser water system in Housekeeping Worked Example 11-2. Assume that half the change in performance was because of condenser cleaning.

$$\text{Reduced electrical costs} = \frac{\$6.705}{2}$$

$$= \$3{,}353$$

An automatic water treatment system was provided for the cooling tower, to optimize water make-up and blowdown, and automatically feed chemicals to control fouling. Capital cost was $3,000. Annual chemical costs are estimated at $800. Note that the system must be cleaned before automatic water treatment is initiated.

$$\text{Simple payback} = \frac{\$3{,}000}{\$3{,}353} = 0.9 \text{ years (11 months)}$$

At the end of the first year, the cost of cleaning the exchangers, the cooling tower, and providing condenser water treatment is negligible. See Housekeeping Worked Example 11-2.

Assuming that the condenser water treatment program is maintained to prevent fouling, annual savings and expenses in following years are:

Electrical savings	$3,353
Less: cost of chemicals	800
Annual savings	$2,553

Other costs are reduced. Annual cleaning of exchangers is eliminated and controlled blowdown reduces make-up water requirements.

2. Heat Pump Versus Electric Heat

A small office addition is planned for an industrial facility in Toronto. An economical means of heating and cooling the addition is desirable. The plant rejects waste heat in the form of warm water. Loads for the proposed building, including ventilation, are 35.17 kW cooling, and 29.31 kW heating. A rooftop packaged air conditioning system with electric heating is proposed. The estimated annual heating cost for the all-electric system is $2,451.

A water-to-air heat pump was considered as an alternative to the basic, air-conditioning with electric heat, rooftop package initially proposed. The heat pump was selected to meet the design heating and cooling loads, with electric duct heaters for 100 per cent backup. The COP for heating at the given water condition was 2.25 and similar to the air-conditioner performance in the summer. The source of warm water was available 85 per cent of the time during the heating season, and cooling water was available throughout the cooling season.

$$\text{Annual heat pump energy costs} = \frac{(0.85 \times 2{,}451)}{2.25} + (0.15 \times 2{,}451)$$

$$= \$1{,}294$$

$$\text{Annual savings} = \$2{,}451 - \$1{,}294$$

$$= \$1{,}157$$

The extra cost for a heat pump package over standard air conditioning with electric heat is estimated at $3,000.

$$\text{Simple payback} = \frac{\$3{,}000}{\$1{,}157} = 2.6 \text{ years}$$

3. Hot Gas Bypass

A small manufacturing plant has a 90 kW capacity refrigeration plant operating at a COP of 3. The compressor has six cylinders and operates at full-load 24 hours per day, 5 days per week and 50 weeks per year. During weekends the refrigeration load is less than 10 per cent of full-load, and the unit uses hot gas bypass to avoid low suction pressures and evaporator frosting. It is proposed to eliminate hot gas bypass and cycle the unit on and off to meet the low loads. Controls will be modified to eliminate hot gas bypass and install anti-short cycle timers at a cost of $1,400. The hot gas bypass imposes a cooling load of about 33 per cent on the unit at a cost of $1,188 per year, in addition to the cost of providing the 9 kW cooling load. By eliminating hot gas bypass, this $1,188 can be saved.

$$\text{Simple payback} = \frac{\$1{,}400}{\$1{,}188} = 1.2 \text{ years}$$

Refer to sample Worksheet 11-1 for detailed calculations.

Retrofit Opportunities

Implemented retrofit opportunities are defined as Energy Management actions that are done once, and for which the cost is significant. Many of the opportunities in this category will required detailed analysis by specialists and cannot be examined in detail in this module. Worked examples are provided for some of the listed Energy Management Opportunities, while in other cases there is only commentary. The following are typical Energy Management Opportunities in the Retrofit category.

- Absorption equipment can provide low cost cooling if dependable, high grade waste heat is available.
- Use a heat pump to upgrade the low temperature waste heat to a temperature suitable for building heating.
- Provide a thermal storage system to reduce compressor cycling, and allow continuous operation at full-load and higher efficiency. See Thermal Storage, Module 19.
- Provide decentralized systems to match loads with specialized requirements. For example, if a large system operates at a low evaporator temperature when only a small portion of the load requires low temperature, provide a small, low temperature system to serve the special area. Operate the large system at a higher evaporator temperature to improve COP. Consider "piggybacking" the low temperature system onto the higher temperature system to reduce temperature differences and increase COP.
- Reclaim rejected condenser heat for space heating, process heating or water preheating. In addition to reclaiming the otherwise wasted heat, the system COP may be increased when a lower temperature condensing medium is available. For example, preheating domestic water will reduce the energy required for water heating and reduce the condensing temperature. The cold incoming water supply can often reduce the condensing water temperature by 5 to 10°C, thereby increasing the system COP. Another example is using condenser heat for melting snow, or underslab heating at an ice rink to reduce frost penetration into the underlying soil.

- Desuperheat the refrigerant vapor (hot gas) leaving the compressor. The superheat can be recovered for process or make-up water preheating. Because the temperature of the hot gas is higher than the condensing temperature, the superheat can be used where lower temperature latent heat cannot. Care must be taken in the design of the refrigerant piping system to ensure proper return of liquid refrigerant and oil from the desuperheater.
- Use well, river or lake water as a lower temperature cooling medium to reduce condensing temperatures. If an air-cooled condenser requires major repair or replacement, consider using an evaporative condenser. Improved performance and reduced energy cost because of the higher COP may justify the added expenditure.
- Use mechanical refrigeration equipment in facilities, such as indoor swimming pools, where high ventilation rates are required for humidity control. Winter heating costs for the ventilation air can be reduced by reducing the ventilation rate. The total heat of rejection can be used to preheat the ventilation supply air and preheat the make-up water for the pool. Energy savings result.

Retrofit Worked Examples

1. Provide Absorption Cooling Equipment

A plant has a dependable, low cost source of steam which is used to operate a steam turbine. A cooling tower is used to produce cooling water for condensing the turbine steam exhaust. It is proposed to install an absorption chiller in the exhaust to generate cooling from the heat otherwise rejected to the cooling tower. Refer to Figures 47 and 48 for the existing and proposed systems.

The turbine exhaust flow-rate is 4000 kg/h of 100 kPa(gauge) saturated steam. Assuming that the piping heat losses are negligible, the heat energy available from condensing the steam is calculated by multiplying the mass-flow rate of steam by the latent heat of steam at 100 kPa(gauge). The latent heat of the steam is obtained from steam tables, and is equal to the enthalpy of the vapor minus the enthalpy of the liquid. The energy available (E_{avail}), can be calculated.

$$E_{avail} = \text{mass flow rate x } (h_g - h_f)$$

$$= 4000 \text{ kg/h x } (2675.4 - 417.5 \text{ kJ/kg})$$

$$= 9.032 \text{ x } 10^6 \text{ kJ/h, or}$$

$$= \frac{9.032 \text{ x } 10^6 \text{ kJ/h}}{3.6 \text{ x } 10^3 \text{ kJ/kWh}} = 2.51 \text{ x } 10^3 \text{ kW}$$

For an absorption refrigeration machine with average COPs the following can be produced at little cost.

For cooling: COP = 0.6

Capacity = 0.6 x 2.51 x 10^3 = 1506 kW

For heating: COP = 1.3

Capacity = 1.3 x 2.51 x 10^3 = 3260 kW

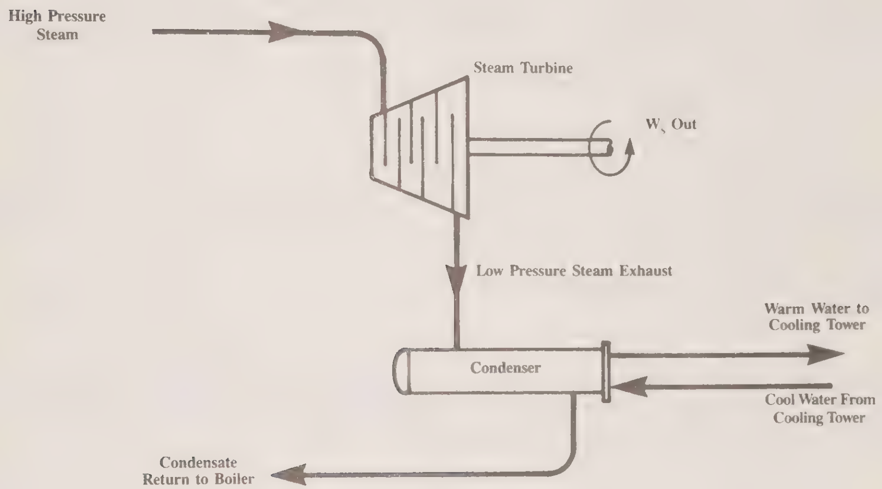
The energy required in this example is provided by waste heat which would otherwise have been rejected. The only energy costs incurred are for operating the pump systems. The energy cost saved by reclaiming the condenser heat is the cost of the energy required to produce the same amount of cooling using conventional (centrifugal) chilling equipment.

The COP for a centrifugal chiller is 4.0 and there will be 2000 hours of full-load operation per year. Approximate annual cooling energy costs avoided by using absorption chillers rather than centrifugal chillers are calculated based on the cost of operating a centrifugal machine.

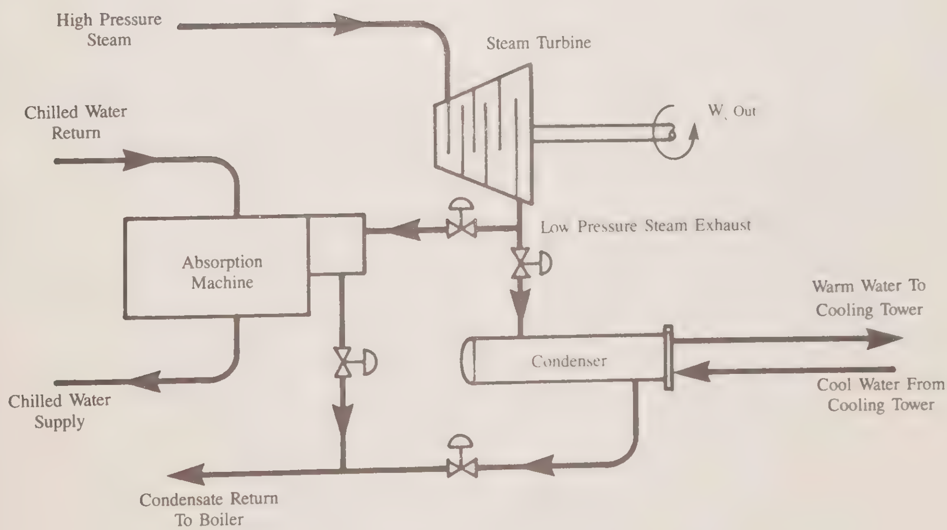
$$\text{Savings} = \frac{1506 \text{ kW x } 2000 \text{ h x } \$0.05/\text{kWh}}{4.0 \text{ (centrifugal chiller COP)}} = \$37{,}650$$

A 1506 kW cooling system costs $90,000.

$$\text{Simple payback} = \frac{\$90{,}000}{\$37{,}650} = 2.4 \text{ years}$$



Existing System
Figure 47



Proposed System
Figure 48

2. Condenser Heat Reclaim

An industrial refrigeration plant rejects 700 kW of cooling load plus 150 kW of compressor work through a water cooled condenser. A cooling tower circulating 38 L/s of condenser water is used to reject this heat. The condenser inlet water temperature is maintained at 29°C with an outlet of 39°C. The facility also has a requirement to heat 10 L/s of process water from 10 to 50°C using a gas fired boiler. The energy rejected from the refrigeration equipment can be used to reduce total energy requirements by preheating the process water with a second heat exchanger prior to the water entering the boiler. The condenser water temperature decreases by 5°C through the heat exchanger, and the heat transfer efficiency for both exchangers is 80 per cent. The existing and proposed systems are shown schematically in Figures 49 and 50.

Worksheets 11-1 and 11-2 are used to illustrate the calculations. The amount of energy saved, 0.634 MJ/s, is for the design condition only. Because of fluctuating loads and temperature differences lower than design, the average heat recovered can be substantially less. In this example, the plant operates an estimated 1500 hours per year at full load.

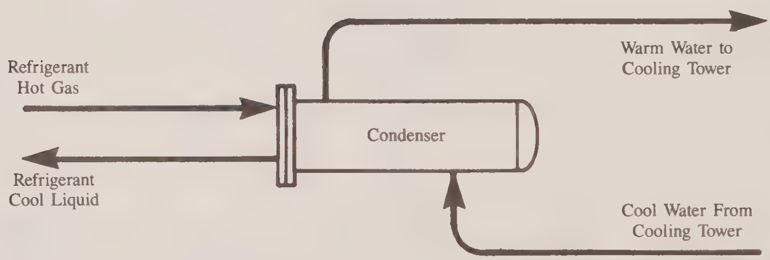
$$\text{Annual energy savings} = 1500 \times 0.634 \times 3600$$

$$= 3.426 \times 10^6 \text{ MJ/yr.}$$

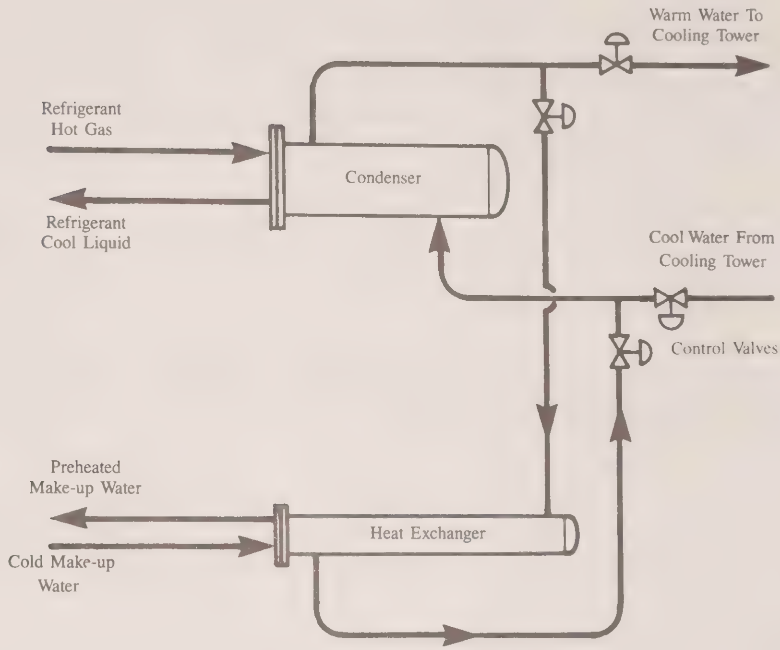
Using Worksheet 11-1, the energy displaced by heat recovery for a gas-fired boiler system will be $16,080 per year. The cost of the heat reclaim system is $45,000.

$$\text{Simple payback} = \frac{\$45{,}000}{\$16{,}080} = 2.8 \text{ years}$$

Other benefits of condenser heat reclaim include make-up water and fan motor power savings for the cooling tower. Where high temperatures are required for process heating, consider *desuperheating* the higher temperature superheated gases instead of extracting lower temperature condenser heat.



Existing Condenser System
Figure 49



Proposed Heat Reclaim System
Figure 50

3. Decentralize Systems for Special Requirements

A refrigeration system provides 4.4°C chilled water for process and air-conditioning loads, 120 hours per week, 50 weeks per year. Total cooling load is 108 kW, of which 4 kW is for an air-conditioning load that requires 4.4°C water for dehumidification. It is proposed to provide a small water cooled air-conditioner dedicated to air-conditioning so that the process cooling water temperature can be raised to 7.8°C, thus raising the COP of the larger chiller system. Both the existing and proposed systems will use the existing cooling tower for heat rejection. Refer to Figures 51 and 52 for details of existing and proposed arrangements.

From manufacturer's catalogue data, the chiller motor power is 26.3 kW at 108 kW cooling load, 4.4°C evaporator and 35°C condensing temperatures. At a 7.8°C evaporator temperature, the motor power is 27.0 kW for a cooling capacity of 119.2 kW. COPs were calculated:

$$\text{COP (at 4.4°C)} = \frac{108}{26.3} = 4.10$$

$$\text{COP (at 7.8°C)} = \frac{119.2}{27} = 4.23$$

The new chiller power required is the reduced cooling load divided by the new COP.

$$\text{Power required} = \frac{(108 - 4)\text{kW}}{4.23} = 24.57 \text{ kW}$$

Using Worksheet 11-1 (A), the annual electricity costs are calculated for the chiller under both operating conditions. The required power input is 1.20 kW for the new air-conditioning system and 0.12 kW for the pump. Capital costs are estimated as $4,000. Using Worksheet 11-1 (B), the energy costs for the new air-conditioning system are calculated. The net annual savings can be obtained.

Savings from chiller COP increase	=	$519
Electrical costs for new air-conditioning system	=	396
Annual savings	=	$123

$$\text{Simple payback} = \frac{\$4{,}000}{\$123} = 32 \text{ years}$$

The calculated payback of this improved COP does not justify making the proposed modifications. However, the same plant operating under different conditions may have a shorter payback period. Techniques used to improve the operating COP by decentralizing systems can produce other attractive energy savings from less obvious sources. For example, consider the same plant and proposed modifications operating under a different schedule.

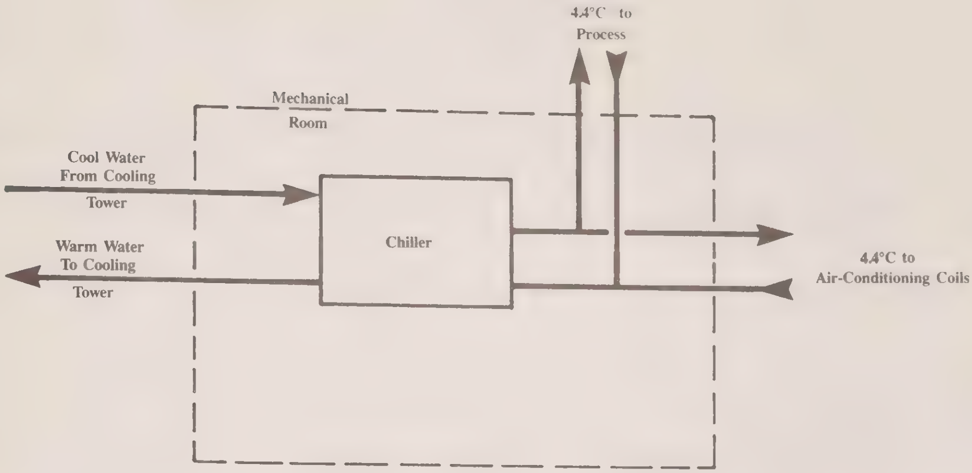
- Process loads: 60 h/week and 50 weeks/yr
- Air-conditioning: 168 h/week and 50 weeks/yr

In order to provide the 4.4°C chilled water required for air-conditioning, the chiller would use hot gas bypass for 108 hours per week. For a six cylinder compressor, the bypass cooling load is about 33 per cent of the nominal cooling capacity, or 108 kW x 0.33. The energy consumed when hot gas bypass is operating is the bypass load, plus the air-conditioning load, divided by the existing COP. This equals 52,683 kWh per year.

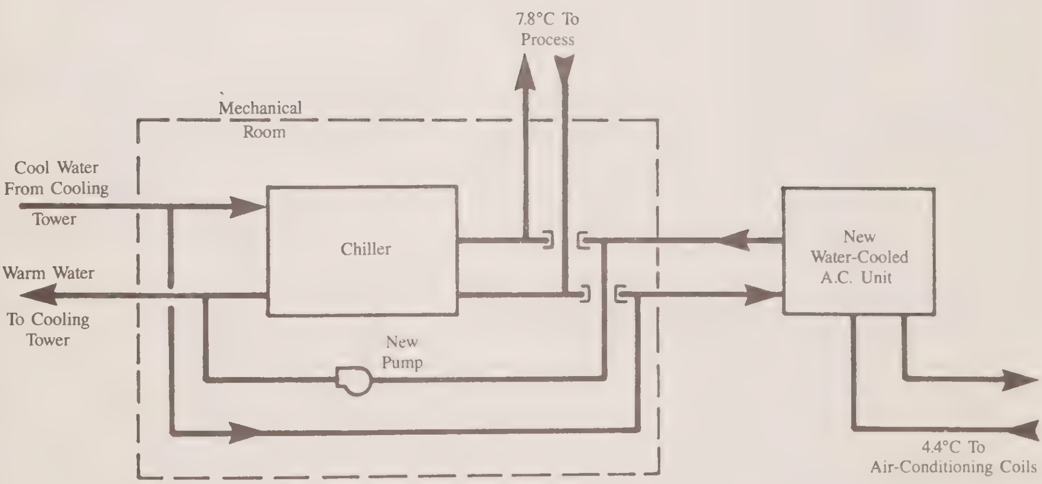
The cost for this energy calculated from Worksheet 11-1 (C) is $2,634 per year. For the new system, the energy cost calculated from Worksheet 11-1 (D), and incurred during the 108 hours per week the chiller is not operating is $356 per year. Net annual energy savings obtained by providing a decentralized system for specific loads is $2,634 − $356 = $2,278 per year.

$$\text{Simple payback} = \frac{\$4{,}000}{\$2{,}278} = 1.8 \text{ years}$$

Reduced cooling tower operation, improved COP operating flexibility and additional cooling capacity obtained for process cooling, are additional benefits not considered in the payback calculations.



Existing Chiller System
Figure 51



Proposed Decentralized System
Figure 52

Energy Costs
Worksheet 11-1

Company: WORKED EXAMPLE 3 Date: 85/4/1

Location: LOW COST By: MBE

Energy ~~Used~~ or Displaced Per Year = ______ MJ/yr

Details:

50 WKS × 48 HR/WK = 2400 HR/YR

or, = ______ MJ/yr / 3.6 MJ/kWh

WHERE, COOLING CAPACITY = 90 kW
HOT GAS BY PASS = 0.33
COP = 3

23760 kWh/yr (1)

$$2400 \times \left(\frac{0.33 \times 90}{3}\right) = 23760$$

Fuel Costs:

- Gas . . . . ($/m³) ______ (2)
- Oil . . . . ($/L) ______ (3)
- Electricity . . . . ($/kWh) 0.05 (4) (Ignore demand charges)
- Other . . . . ($/unit) ______ (5) ______ unit (L,m³,tonne,kg)

Energy content of fuels (other than electricity)

- Fuel type ______
- Energy content ______ MJ per ______ (L,kg,m³,tonne) (6)

(From plant data or Appendix C)

Energy Costs

Worksheet 11-1

Page 2 of 2

Company: WORKED EXAMPLE 3 Date: 85/4/1

Location: LOW COST By: MBE

Cost of energy ~~used~~ or displaced per year

- Gas

$$AG = \frac{\_\_\_\_\_\_ (1) \times \_\_\_\_\_\_ (2)}{\_\_\_\_\_\_ (6) \times \_\_\_\_\_\_ (\text{eff.})} = \$\_\_\_\_\_\_/\text{yr}$$

- Gas

$$AG = \frac{\_\_\_\_\_\_ (1) \times \_\_\_\_\_\_ (2)}{\_\_\_\_\_\_ (6) \times \_\_\_\_\_\_ (\text{eff.})} = \$\_\_\_\_\_\_/\text{yr}$$

- Oil

$$AO = \frac{\_\_\_\_\_\_ (1) \times \_\_\_\_\_\_ (3)}{\_\_\_\_\_\_ (6) \times \_\_\_\_\_\_ (\text{eff.})} = \$\_\_\_\_\_\_/\text{yr}$$

- Electricity

$$AE = 23760 \ (1) \times 0.05 \ (4) = \$ \ 1188 \ /\text{yr}$$

- Other

$$AF = \frac{\_\_\_\_\_\_ (1) \times \_\_\_\_\_\_ (5)}{\_\_\_\_\_\_ (6) \times \_\_\_\_\_\_ (\text{eff.})} = \$\_\_\_\_\_\_/\text{yr}$$

Where, eff. = efficiency of heating system

AG = annual cost of gas

AO = annual cost of oil

AE = annual cost of electricity

AF = annual cost of fuel

For gas and oil fired systems, eff. = 0.8

For electric systems, eff. = 1.0

Heat Energy Available From Waste Water Stream
To Preheat A Water Stream
(Approximate Method)
Worksheet 11-2
Page 1 of 2

Company: WORKED EXAMPLE 2 Date: 85/3/18

Location: RETROFIT By: MBE

Waste Water Stream CONDENSER WATER

- Water flow (fw) 38.0 L/s
- Present water temperature (t_1) 39.0 °C
- Proposed water leaving temperature (t_2) 34.0 °C
  (Input of heat exchanger manufacturer required to establish this figure, or estimate using $t_2 = t_1 - 5°$)
- Heat available, (Q) = fw x ($t_1 - t_2$) x 15 MJ/h

= 38.0 x (39.0 − 34.0) x 15

= 2850 MJ/h

or $\dfrac{2850}{3600} \dfrac{\text{MJ/h}}{\text{s/h}}$

= 0.792 MJ/s

Proposed heat exchanger efficiency (from heat exchanger manufacturer) 80 %

Heat available = 0.792 MJ/s x $\dfrac{80\ \%}{100}$

= 0.633 MJ/s (1)

Heat Energy Available From Waste Water Stream
To Preheat A Water Stream
(Approximate Method)
Worksheet 11-2
Page 2 of 2

Company: WORKED EXAMPLE 2 Date: 85/3/18

Location: RETROFIT By: MBE

Process Stream

Water flow (fw) 10.0 L/s

Entering water temperature (t_1) 10.0 °C

Required water temperature (t_2) 50.0 °C

Heat required, (Q) = fw x (t_1 − t_2) x 15MJ/h

= 10 x (10 − 50.0) x 15

= − 6000 MJ/h

or $\frac{6000}{3600}$ $\frac{\text{MJ/h}}{\text{s/h}}$ (drop negative sign)

= 1.667 MJ/s (2)

Heat energy required for final heating of process water stream

= 1.667 (2) − 0.633 (1)

= 1.034 MJ/s (3)

Energy Costs
Worksheet 11-1
Page 1 of 2

Company: WORKED EXAMPLE 2 Date: 85/3/18

Location: RETROFIT By: MBE

Energy Used or Displaced Per Year = 3 418 200 MJ/yr

Details:
0.633 X 3600 X 1500
WHERE, 0.633 = MJ/S
3600 = S/HR
1500 = HR/YR

or, = $\frac{3\ 418\ 200}{3.6}$ $\frac{\text{MJ/yr}}{\text{MJ/kWh}}$

= 949 500 kWh/yr (1)

Fuel Costs:

- Gas . . . . ($/m³) 0.14 (2)
- Oil . . . . ($/L) ________ (3)
- Electricity . . . . ($/kWh) ________ (4) (Ignore demand charges)
- Other . . . . ($/unit) ________ (5) ________ unit (L,m³,tonne,kg)

Energy content of fuels (other than electricity)

- Fuel type NATURAL GAS
- Energy content 37.2 MJ per m^3 (L,kg,m³,tonne) (6)

(From ~~plant data~~ or Appendix C)

Energy Costs

Worksheet 11-1

Company: WORKED EXAMPLE 2 Date: 85/3/18

Location: RETROFIT By: MBE

Cost of energy ~~used~~ or displaced per year

- Gas

$$AG = \frac{3\,418\,200\ (1) \times 0.14\ (2)}{37.2\ (6) \times 0.8\ (\text{eff.})} = \$\ 16\,080\ /\text{yr}$$

- Gas

$$AG = \frac{\_\_\_\ (1) \times \_\_\_\ (2)}{\_\_\_\ (6) \times \_\_\_\ (\text{eff.})} = \$\_\_\_\ /\text{yr}$$

- Oil

$$AO = \frac{\_\_\_\ (1) \times \_\_\_\ (3)}{\_\_\_\ (6) \times \_\_\_\ (\text{eff.})} = \$\_\_\_\ /\text{yr}$$

- Electricity

$$AE = \_\_\_\ (1) \times \_\_\_\ (4) = \$\_\_\_\ /\text{yr}$$

- Other

$$AF = \frac{\_\_\_\ (1) \times \_\_\_\ (5)}{\_\_\_\ (6) \times \_\_\_\ (\text{eff.})} = \$\_\_\_\ /\text{yr}$$

Where, eff. = efficiency of heating system

AG = annual cost of gas

AO = annual cost of oil

AE = annual cost of electricity

AF = annual cost of fuel

For gas and oil fired systems, eff. = 0.8

For electric systems, eff. = 1.0

Energy Costs
Worksheet 11-1 (A)
Page 1 of 2

Company: WORKED EXAMLE 3 Date: 85/3/18

Location: RETROFIT By: MBE

Energy ~~Used~~ or Displaced Per Year = ______________ MJ/yr

Details:
120 HR/WK OPERATION
50 WK/YR
ORIGINAL CHILLER KW=26.3
REVISED CHILLER kW=24.57
120×50×(26.3−24.57)= 10 380

or, = ______________ MJ/yr
3.6 MJ/kWh

= 10 380 kWh/yr (1)

Fuel Costs:

- Gas . . . . ($/$m^3$) ______________ (2)
- Oil . . . . ($/L) ______________ (3)
- Electricity . . . . ($/kWh) 0.05 (4) (Ignore demand charges)
- Other . . . . ($/unit) __________ (5) __________ unit (L,m^3,tonne,kg)

Energy content of fuels (other than electricity)

- Fuel type ______________
- Energy content ______________ MJ per ______________ (6)
  (L,kg,m^3,tonne)

(From plant data or Appendix C)

Energy Costs
Worksheet 11-1 (A)

Company: WORKED EXAMPLE 3 Date: 85/3/18

Location: RETROFIT By: MBE

Cost of energy used or displaced per year

- Gas

AG = $\dfrac{\_\_\_\_\ (1) \times \_\_\_\_\ (2)}{\_\_\_\_\ (6) \times \_\_\_\_\ (\text{eff.})}$ = $____ /yr

- Gas

AG = $\dfrac{\_\_\_\_\ (1) \times \_\_\_\_\ (2)}{\_\_\_\_\ (6) \times \_\_\_\_\ (\text{eff.})}$ = $____ /yr

- Oil

AO = $\dfrac{\_\_\_\_\ (1) \times \_\_\_\_\ (3)}{\_\_\_\_\ (6) \times \_\_\_\_\ (\text{eff.})}$ = $____ /yr

- Electricity

AE = 10 380 (1) x 0.05 (4) = $ 519 /yr

- Other

AF = $\dfrac{\_\_\_\_\ (1) \times \_\_\_\_\ (5)}{\_\_\_\_\ (6) \times \_\_\_\_\ (\text{eff.})}$ = $____ /yr

Where, eff. = efficiency of heating system

AG = annual cost of gas

AO = annual cost of oil

AE = annual cost of electricity

AF = annual cost of fuel

For gas and oil fired systems,eff. = 0.8

For electric systems, eff. = 1.0

Energy Costs
Worksheet 11-1 (B)
Page 1 of 2

Company: WORKED EXAMPLE 3 Date: 85/3/18

Location: RETROFIT By: MBE

Energy Used or ~~Displaced~~ Per Year = ____________ MJ/yr

Details:
PUMP kW = 0.12
AIR CONDITIONER kW=1.20
120 HR/WK
50 WK/YR
120 x 50 x (1.20+0.12)

or, = $\dfrac{\text{\_\_\_\_\_\_\_\_ MJ/yr}}{3.6 \text{ MJ/kWh}}$

= 7 920 kWh/yr (1)

Fuel Costs:

- Gas . . . . ($/m^3) ____________(2)
- Oil . . . . ($/L) ____________(3)
- Electricity . . . . ($/kWh) 0.05 (4) (Ignore demand charges)
- Other . . . . ($/unit) ________ (5) ________ unit (L,m^3,tonne,kg)

Energy content of fuels (other than electricity)

- Fuel type ____________________
- Energy content ____________ MJ per ____________ (L,kg,m^3,tonne) (6)

(From plant data or Appendix C)

Energy Costs
Worksheet 11-1 (B)
Page 2 of 2

Company: WORKED EXAMPLE 3 Date: 85/3/18

Location: RETROFIT By: MBE

Cost of energy used or ~~displaced~~ per year

- Gas

$$AG = \frac{\_\_\_\_\_\_\_\_\ (1) \times \_\_\_\_\_\_\_\_\ (2)}{\_\_\_\_\_\_\_\_\ (6) \times \_\_\_\_\_\_\_\_\ (\text{eff.})} = \$\_\_\_\_\_\_\_\_ /\text{yr}$$

- Gas

$$AG = \frac{\_\_\_\_\_\_\_\_\ (1) \times \_\_\_\_\_\_\_\_\ (2)}{\_\_\_\_\_\_\_\_\ (6) \times \_\_\_\_\_\_\_\_\ (\text{eff.})} = \$\_\_\_\_\_\_\_\_ /\text{yr}$$

- Oil

$$AO = \frac{\_\_\_\_\_\_\_\_\ (1) \times \_\_\_\_\_\_\_\_\ (3)}{\_\_\_\_\_\_\_\_\ (6) \times \_\_\_\_\_\_\_\_\ (\text{eff.})} = \$\_\_\_\_\_\_\_\_ /\text{yr}$$

- Electricity

$$AE = 7920\ (1) \times 0.05\ (4) = \$\ 396 /\text{yr}$$

- Other

$$AF = \frac{\_\_\_\_\_\_\_\_\ (1) \times \_\_\_\_\_\_\_\_\ (5)}{\_\_\_\_\_\_\_\_\ (6) \times \_\_\_\_\_\_\_\_\ (\text{eff.})} = \$\_\_\_\_\_\_\_\_ /\text{yr}$$

Where, eff. = efficiency of heating system

AG = annual cost of gas

AO = annual cost of oil

AE = annual cost of electricity

AF = annual cost of fuel

For gas and oil fired systems,eff. = 0.8

For electric systems, eff. = 1:0

Energy Costs
Worksheet 11-1 (C)
Page 1 of 2

Company: WORKED EXAMPLE 3 Date: 85/3/18

Location: RETROFIT By: MBE

Energy Used or ~~Displaced~~ Per Year = ______ MJ/yr

Details:
HR/YR = 108 × 50 = 5400 or, = ______ MJ/yr
COP = 4.1 3.6 MJ/kWh
REFRIGERATION LOAD = BYPASS PLUS
COOLING LOAD = (0.33 × 108) + 4 = 52 683 kWh/yr (1)

∴ ANNUAL ENERGY

$$= 5400 \times \frac{(0.33 \times 108 + 4)}{4.1} = 52\,683$$

Fuel Costs:

- Gas . . . . ($/m^3) ______ (2)
- Oil . . . . ($/L) ______ (3)
- Electricity . . . . ($/kWh) 0.05 (4) (Ignore demand charges)
- Other . . . . ($/unit) ______ (5) ______ unit (L,m^3,tonne,kg)

Energy content of fuels (other than electricity)

- Fuel type ______
- Energy content ______ MJ per ______ (6)
  (L,kg,m^3,tonne)

(From plant data or Appendix C)

Energy Costs
Worksheet 11-1 (c)
Page 2 of 2

Company: WORKED EXAMPLE 3 Date: 85/3/18

Location: RETROFIT By: MBE

Cost of energy used or ~~displaced~~ per year

- Gas

$$AG = \frac{\_\_\_\_\_ (1) \times \_\_\_\_\_ (2)}{\_\_\_\_\_ (6) \times \_\_\_\_\_ (\text{eff.})} = \$\_\_\_\_\_ /\text{yr}$$

- Gas

$$AG = \frac{\_\_\_\_\_ (1) \times \_\_\_\_\_ (2)}{\_\_\_\_\_ (6) \times \_\_\_\_\_ (\text{eff.})} = \$\_\_\_\_\_ /\text{yr}$$

- Oil

$$AO = \frac{\_\_\_\_\_ (1) \times \_\_\_\_\_ (3)}{\_\_\_\_\_ (6) \times \_\_\_\_\_ (\text{eff.})} = \$\_\_\_\_\_ /\text{yr}$$

- Electricity

$$AE = 52\,683\ (1) \times 0.05\ (4) = \$\ 2364\ /\text{yr}$$

- Other

$$AF = \frac{\_\_\_\_\_ (1) \times \_\_\_\_\_ (5)}{\_\_\_\_\_ (6) \times \_\_\_\_\_ (\text{eff.})} = \$\_\_\_\_\_ /\text{yr}$$

Where, eff. = efficiency of heating system

AG = annual cost of gas

AO = annual cost of oil

AE = annual cost of electricity

AF = annual cost of fuel

For gas and oil fired systems,eff. = 0.8

For electric systems, eff. = 1.0

Energy Costs
Worksheet 11-1 (D)
Page 1 of 2

Company: WORKED EXAMPLE 3 Date: 85/3/18

Location: RETROFIT By: MBE

Energy Used or ~~Displaced~~ Per Year = ______ MJ/yr

Details:
PUMP kW = 0.12
AIR CONDITIONER kW = 1.20
108 HR/WK
50 WK/YR
108 x 50 x (1.20 + 0.12) = 7128

or, = ______ MJ/yr
3.6 MJ/kWh

= 7128 kWh/yr (1)

Fuel Costs:

- Gas . . . . ($/m³) ______ (2)
- Oil . . . . ($/L) ______ (3)
- Electricity . . . . ($/kWh) 0.05 (4) (Ignore demand charges)
- Other . . . . ($/unit) ______ (5) ______ unit (L,m³,tonne,kg)

Energy content of fuels (other than electricity)

- Fuel type ______
- Energy content ______ MJ per ______ (6)
  (L,kg,m³,tonne)

(From plant data or Appendix C)

Energy Costs
Worksheet 11-1 (D)

Company: WORKED EXAMPLE 3 Date: 85/3/18

Location: RETROFIT By: MBE

Cost of energy used or ~~displaced~~ per year

- Gas

$$AG = \frac{\_\_\_\_\_\_ (1) \times \_\_\_\_\_\_ (2)}{\_\_\_\_\_\_ (6) \times \_\_\_\_\_\_ (\text{eff.})} = \$\_\_\_\_\_\_/yr$$

- Gas

$$AG = \frac{\_\_\_\_\_\_ (1) \times \_\_\_\_\_\_ (2)}{\_\_\_\_\_\_ (6) \times \_\_\_\_\_\_ (\text{eff.})} = \$\_\_\_\_\_\_/yr$$

- Oil

$$AO = \frac{\_\_\_\_\_\_ (1) \times \_\_\_\_\_\_ (3)}{\_\_\_\_\_\_ (6) \times \_\_\_\_\_\_ (\text{eff.})} = \$\_\_\_\_\_\_/yr$$

- Electricity

$$AE = 7128\ (1) \times 0.05\ (4) = \$\ 356\ /yr$$

- Other

$$AF = \frac{\_\_\_\_\_\_ (1) \times \_\_\_\_\_\_ (5)}{\_\_\_\_\_\_ (6) \times \_\_\_\_\_\_ (\text{eff.})} = \$\_\_\_\_\_\_/yr$$

Where, eff. = efficiency of heating system

AG = annual cost of gas

AO = annual cost of oil

AE = annual cost of electricity

AF = annual cost of fuel

For gas and oil fired systems, eff. = 0.8

For electric systems, eff. = 1.0

APPENDICES

A Glossary
B Tables
C Conversion Factors
D Worksheets

Glossary

Absolute Pressure — Any pressure where the base for measurement is full vacuum. Expressed in kPa (absolute).

Absorber — A device containing liquid solvent for absorbing refrigerant (or other) vapors. In an absorption refrigeration system, it is the low pressure vessel where refrigerant vapor is absorbed by the solvent.

Affinity — The property of attraction by which different substances, when brought into contact with each other, unite to form a new compound, as in the affinity between lithium bromide and water in an absorption refrigeration system.

ANSI — American National Standards Institute.

Approach — In a heat exchanger, the difference in temperature between the leaving treated fluid and the entering working fluid. In an evaporative cooling device, the difference between the average temperature of the leaving circulating water and the average wet bulb temperature of the leaving air.

Aquifer — A porous geological formation containing water.

ASHRAE — The American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc.

Audit, diagnostic — The analysis of a potential opportunity to save energy which could involve the assessment of the current process operation and records, and the calculation of savings, of capital and operating costs so that the financial viability of the project can be established.

Audit, walk through — The visual inspection of a facility to observe how energy is used or wasted.

Brine — Any liquid used for the transfer of heat without a change of state, having no flash point, or a flash point above 150°F, as determined by ASTM method D93.

Capillary Tube — In refrigeration practice, a tube of small internal diameter used as a refrigerant flow control or expansion device between the high and low pressure side of the circuit. Also used to transmit pressure from the sensing bulb of some temperature controls to the operating element.

Chiller — Packaged refrigeration equipment specifically designed to produce chilled water or brine. Generally consists of liquid to refrigerant heat exchangers for both evaporator and condenser, centrifugal compressor, throttling devices and controls.

Chiller, double bundle — A chiller with dual condensers so that heat can be rejected to either a cooling tower or a heat reclaim system, or both.

Clearance Volume — The space in a cylinder not occupied by the piston at the end of the compression stroke, measured in percent of piston displacement.

Compression Ratio — The ratio of discharge pressure to suction pressure of a compressor expressed in absolute units.

Concentrator — See generator

Condenser — A heat exchanger in which the refrigerant, after having been compressed to a suitable pressure, is condensed by rejection of heat to an external cooling medium.

Condensation — The process of changing a vapor into liquid by the extraction of heat.

Cooling Water — Water used for cooling a process or condenser.

Cycle, closed — Any cycle in which the primary medium is always enclosed and repeats the same sequence of events.

Degree Days — A measure to correlate the outside temperature with the energy required for heating based on the assumption that heating is required when the average daily temperature is less than 18°C. Thus, one heating Degree Day results from an average of 17°C for 24 hours. The Degree Days are accumulated to provide a means of comparing the influence of weather on energy usage over a period of time.

Degree Kelvin — A unit of temperature measurement where zero Kelvin (0 K) is *absolute to zero* and is equal to -273°C. The K and °C are equal increments of temperature.

Drift — In a cooling tower or spray pond, is the amount of warm water lost in the form of spray or fine mist, carried along by the wind or draft at the unit.

Ejector, steam — A nozzle-like device that builds up a high fluid velocity in a restricted area to obtain a lower static pressure at that point so that fluid from another source can be drawn in.

Energy — The capacity for doing work; taking a number of forms that may be transformed from one into another, such as thermal (heat), mechanical (work), electrical, and chemical; in customary units, measured in kilowatt-hours (kWh) or megajoules (MJ).

Energy Efficiency Ratio (EER) — A measure of efficiency of a refrigeration or heat pump system comparing the actual refrigeration effect to the actual work or energy input to the cycle. (Note:This ratio exists only in the imperial form of measurement.)

Energy, waste — Energy which is lost without being fully utilized. It may include the energy in a fluid or stream such as steam, exhaust gas, discharge water or even refuse.

Enthalpy — Enthalpy is a measure of the heat energy per unit mass of a material. Units are expressed as kJ/kg.

Entropy — The ratio of the heat added to a substance to the absolute temperature at which it was added; a measure of the molecular disorder of a substance at a given state.

Evaporator — A heat exchanger in which the liquid refrigerant, is evaporated by absorption of heat from the medium to be cooled.

Flash Chamber — A separating tank placed between the expansion valve and the evaporator to separate and bypass any flash gas formed in the expansion valve.

Flash Gas — The gas resulting from the instantaneous evaporation of a liquid when the pressure above the liquid is reduced. The remaining liquid is thereby cooled. Generation of flash gas occurs at the throttling device in a refrigeration process when the pressure above the liquid is reduced to below the saturation pressure of the liquid at a given temperature.

Fusion — Melting of a substance.

Gas, dry saturated — Vapor containing no liquid, whose quality is one(1); and whose state can be graphically defined as being on the saturation line of a state diagram, to the right of the critical point.

Gas, hot — The vapor discharged from the compressor in a vapor compression cycle - the gas is "hot' as compared to the suction gas due to the addition of energy in the form of work of compression.

Gas, suction — The refrigerant vapor on the suction side of a compressor in a vapor compression cycle; the evaporator coil discharge gas.

Gauge Pressure — Any pressure where the base for measurement is atmospheric pressure expressed as kPa(gauge). Note kPa(gauge) + atmospheric pressure = kPa (absolute).

Generator — A basic part of an absorption refrigeration system. A still provided with means of heating to drive refrigerant out of solution.

Heat Exchanger — A device specifically designed to transfer heat between two physically separated fluids.

Heat Sink — The medium or device to which heat is rejected; must always be at a lower temperature than that of the medium rejecting heat for heat transfer to occur. For example a condenser rejects heat to a lake through a heat exchanger - the lake acts as a heat sink.

Heat Source — A medium or process from which heat may be extracted; must always be at a temperature higher than the medium or process requiring the heat for heat transfer to occur; see waste heat. For example, flash steam from a boiler plant might be used as heat source for regenerating the refrigerant from the solvent in an absorption chiller.

Heat, waste — Energy in the form of heat rejected or lost from a process, which may be recovered or reused in another process providing it is of sufficient quality (i.e. hot enough and there is a use for it).

Helical — Spiral shaped, as in a screw thread.

Insulation — A material of low thermal conductivity used to reduce the passage of heat.

Intercooling — The removal of heat from compressed gas between compression stages; often accomplished by injecting lower temperature flash gas removed in a flash chamber at the throttling process.

Latent Heat — The change of enthalpy during a change of state; the amount of heat absorbed or rejected at constant temperature at any pressure; or the difference in enthalpies of a pure condensable fluid between its dry saturated state and its saturated liquid state at the same pressure.

Make-up Water -- Water added to a process to replace that lost during the process, for example: by evaporation or drift from a cooling tower or leaks from a chilled water system.

Process — A change in the properties of a system or substance, usually defined by the conditions at the start and at the end of the process.

Property — A physical characteristic of a substance used to describe its state. Any two properties will usually define the state or condition of the substance from which all other properties can be derived. Examples are temperature, pressure, enthalpy and entropy.

Recuperator — A heat exchanger designed to recover what otherwise would be waste heat.

Rectifier — A device used to refine or purify a liquid by repeated distillations until the desired degree of purity is obtained, often a component of the generator in an ammonia absorption cycle machine.

Refrigerating Capacity — The net or nominal cooling or heating capacity of a given refrigeration system, expressed in watts or kilowatts cooling.

Refrigeration Effect — The net quantity of cooling or heat rejection obtained, not including spurious effects such as heat gains from piping or the heat added to the refrigerant vapor at the compressor.

Saturation Pressure — The pressure of a liquid or vapor at saturation conditions (i.e. on or under the saturation curve).

Saturation Temperature — The temperature of a liquid or vapor at saturation conditions (i.e. on or under the saturation curve).

Separator, oil — A device used for separating oil and oil vapor from a refrigerant, usually installed in the compressor discharge line.

Solute — The material which dissolves in the solvent (e.g. salt is the solute in a salt and water solution, and water is the solvent).

Solvent — The liquid which serves to dissolve the solute in a solution (e.g. water is the solvent in an ammonia absorption system. The ammonia is dissolved into the water).

Specific Volume — The volume per unit mass of a substance, usually expressed in cubic metres per kilogram (m^3/kg); the inverse of specific density.

State — The state of a system or substance is the condition of the system or substance characterized by the values of its properties such as temperature and pressure. The term is often used interchangeably with the term "phase", as in the solid phase or gaseous phase of a substance.

Strong Liquor — In an absorption refrigeration system, the concentrated solvent solution containing little solute leaving the generator and returning to the absorber.

Subcooling — The process of cooling a liquid below its condensing temperature for a given pressure.

Thermal Resistance — A number indicating the insulating value or resistance to heat flow of a material or assembly.

Throttling of a Fluid — A process which consists of lowering pressure by expansion without work or heat input.

Ton of Refrigeration — A useful refrigerating effect equal to 12,000 Btu per hr; 1 ton = 3.517 kW of refrigeration.

Vapor — A gas at or near equilibrium with the liquid phase; a gas under the saturation curve or only slightly beyond the saturated vapor line.

Water, condenser — Cool water circulated through a condenser to condense a refrigerant or vapor.

Work — Mechanical energy imparted in a process or to a substance.

Zero, Absolute — The theoretical temperature at which all molecular motion stops, defined as -273°Celsius, or 0 Kelvin.

REFRIGERANT R-12 PROPERTIES OF LIQUID AND SATURATED VAPOR
TABLE 1

Temp	Pressure	Volume Vapor	Density Liquid	Enthalpy Liquid	Enthalpy Vapor	Entropy Liquid	Entropy Vapor
K	MPa	m^3/kg	kg/m^3	kJ/kg	kJ/kg	kJ/kg · K	kJ/kg · K
170	0.000867	13.460	1686.1	328.51	523.56	3.7732	4.9205
175	0.001395	8.6113	1673.2	332.73	525.75	3.7977	4.9006
180	0.002175	5.6757	1660.1	336.96	527.97	3.8215	4.8826
185	0.003298	3.8433	1647.0	341.18	530.21	3.8446	4.8664
190	0.004875	2.6673	1633.8	345.40	532.48	3.8671	4.8517
195	0.007039	1.8930	1620.5	349.63	534.77	3.8891	4.8385
200	0.009948	1.3713	1607.2	353.87	537.07	3.9105	4.8265
205	0.013787	1.0121	1593.7	358.11	539.39	3.9315	4.8157
210	0.018765	0.75991	1580.1	362.37	541.72	3.9520	4.8060
215	0.025118	0.57957	1566.4	366.64	544.05	3.9721	4.7972
220	0.033110	0.44844	1552.6	370.94	546.40	3.9918	4.7893
222	0.036829	0.40624	1547.1	372.66	547.33	3.9996	4.7864
224	0.040876	0.36876	1541.5	374.39	548.27	4.0073	4.7836
226	0.045271	0.33540	1535.9	376.12	549.21	4.0150	4.7809
228	0.050035	0.30565	1530.3	377.85	550.15	4.0226	4.7783
230	0.055189	0.27905	1524.7	379.59	551.09	4.0302	4.7758
232	0.060756	0.25522	1519.0	381.34	552.02	4.0377	4.7734
234	0.066758	0.23383	1513.3	383.08	552.96	4.0452	4.7712
236	0.073218	0.21459	1507.6	384.84	553.90	4.0526	4.7690
238	0.080160	0.19726	1501.9	386.59	554.83	4.0600	4.7669
240	0.087609	0.18161	1496.1	388.36	555.77	4.0674	4.7649
242	0.095589	0.16745	1490.3	390.13	556.70	4.0747	4.7630
243.36	0.101325	0.15861	1486.3	391.33	557.33	4.0797	4.7618
244	0.10413	0.15463	1484.5	391.90	557.63	4.0820	4.7612
246	0.11324	0.14299	1478.6	393.68	558.56	4.0892	4.7595
248	0.12297	0.13241	1472.7	395.46	559.49	4.0964	4.7578
250	0.13334	0.12278	1466.8	397.25	560.42	4.1036	4.7562
252	0.14436	0.11399	1460.9	399.05	561.34	4.1107	4.7547
254	0.15608	0.10597	1454.9	400.86	562.26	4.1178	4.7533
256	0.16852	0.09863	1448.9	402.67	563.18	4.1249	4.7519
258	0.18170	0.09191	1442.8	404.48	564.10	4.1319	4.7506
260	0.19566	0.08374	1436.7	406.31	565.01	4.1389	4.7493
262	0.21042	0.08007	1430.6	408.14	565.92	4.1459	4.7481
264	0.22602	0.07486	1424.4	409.97	566.82	4.1528	4.7470
266	0.24248	0.07005	1418.2	411.82	567.73	4.1598	4.7459
268	0.25983	0.06563	1412.0	413.67	568.62	4.1667	4.7448
270	0.27811	0.06154	1405.7	415.53	569.52	4.1735	4.7438
272	0.29735	0.05775	1399.3	417.40	570.41	4.1804	4.7429
274	0.31757	0.05425	1392.9	419.27	571.29	4.1872	4.7420
276	0.33881	0.05101	1386.5	421.16	572.17	4.1940	4.7411
278	0.36110	0.04800	1380.0	423.05	573.05	4.2007	4.7403
280	0.38448	0.04520	1373.4	424.95	573.91	4.2075	4.7395
282	0.40896	0.04260	1366.8	426.86	574.78	4.2142	4.7388
284	0.43459	0.04018	1360.1	428.77	575.63	4.2209	4.7380
286	0.46140	0.03793	1353.4	430.70	576.48	4.2276	4.7373
288	0.48943	0.03583	1346.6	432.63	577.33	4.2343	4.7367
290	0.51870	0.03386	1339.7	434.58	578.16	4.2409	4.7360
292	0.54924	0.03203	1332.7	436.53	578.99	4.2475	4.7354
294	0.58111	0.03031	1325.7	438.49	579.81	4.2542	4.7348
296	0.61431	0.02870	1318.6	440.46	580.62	4.2608	4.7343
298	0.64890	0.02719	1311.4	442.44	581.42	4.2673	4.7337
300	0.68491	0.02578	1304.2	444.43	582.21	4.2739	4.7332
302	0.72236	0.02445	1296.8	446.43	582.99	4.2804	4.7326
304	0.76131	0.02320	1289.3	448.44	583.76	4.2870	4.7321
306	0.80177	0.02203	1281.8	450.46	584.53	4.2935	4.7316
308	0.84380	0.02092	1274.1	452.49	585.27	4.3000	4.7311
310	0.88742	0.01988	1266.4	454.53	586.01	4.3065	4.7306
315	1.0037	0.01753	1246.5	459.68	587.79	4.3227	4.7294
320	1.1308	0.01549	1225.8	464.91	589.49	4.3388	4.7281
325	1.2693	0.01371	1204.1	470.21	591.07	4.3549	4.7268
330	1.4198	0.01214	1181.5	475.61	592.54	4.3710	4.7253
335	1.5832	0.01077	1157.6	481.10	593.86	4.3871	4.7237
340	1.7600	0.009549	1132.3	486.71	595.02	4.4033	4.7218
345	1.9510	0.008465	1105.4	492.45	595.98	4.4195	4.7196
350	2.1572	0.007493	1076.5	498.36	596.71	4.4360	4.7170
355	2.3794	0.006616	1045.3	504.47	597.14	4.4527	4.7138
360	2.6188	0.005819	1011.1	510.84	597.20	4.4699	4.7098
365	2.8765	0.005085	972.99	517.57	596.77	4.4878	4.7047
370	3.1541	0.004398	929.67	524.81	595.62	4.5066	4.6980
375	3.4532	0.003735	878.34	532.83	593.34	4.5273	4.6887
380	3.7764	0.003048	811.63	542.34	588.78	4.5515	4.6737
*384.95	4.125	0.001792	558.0	566.9	566.9	4.614	4.614

*Critical point

Courtesy of ASHRAE

REFRIGERANT R-12 PROPERTIES OF SUPERHEATED VAPOR
TABLE 2

Temp K	Abs Pressure 0.01 MPa (Sat'n Temp 200.06 K)		
t	1/v	h	s
(Sat'n)	0.73275	537.11	4.8264
210	0.69713	542.05	4.8505
220	0.66470	547.15	4.8742
230	0.63525	552.38	4.8974
240	0.60836	557.71	4.9201
250	0.58370	563.17	4.9424
260	0.56100	568.73	4.9642
270	0.54002	574.40	4.9856
280	0.52058	580.17	5.0066
290	0.50250	586.05	5.0272
300	0.48564	592.02	5.0475
310	0.46989	598.09	5.0674
320	0.45513	604.25	5.0869
330	0.44128	610.49	5.1061
340	0.42825	616.83	5.1250
350	0.41597	623.24	5.1436
360	0.40438	629.73	5.1619
370	0.39341	636.30	5.1799
380	0.38303	642.94	5.1976
390	0.37319	649.65	5.2150
400	0.36383	656.42	5.2322
410	0.35494	663.27	5.2491
420	0.34647	670.18	5.2657
430	0.33840	677.15	5.2821
440	0.33069	684.18	5.2983
450	0.32333	691.26	5.3142
460	0.31629	698.40	5.3299
470	0.30955	705.60	5.3454
480	0.30309	712.84	5.3607
490	0.29690	720.14	5.3757
500	0.29095	727.48	5.3905

Temp K	Abs Pressure 0.04 MPa (Sat'n Temp 223.58 K)		
t	1/v	h	s
(Sat'n)	2.6578	548.08	4.7841
230	2.5778	551.53	4.7994
240	2.4633	556.98	4.8226
250	2.3594	562.52	4.8452
260	2.2644	568.16	4.8673
270	2.1773	573.89	4.8889
280	2.0969	579.72	4.9101
290	2.0225	585.64	4.9309
300	1.9533	591.65	4.9513
310	1.8889	597.75	4.9713
320	1.8287	603.93	4.9909
330	1.7723	610.20	5.0102
340	1.7193	616.55	5.0292
350	1.6694	622.98	5.0478
360	1.6225	629.49	5.0661
370	1.5781	636.07	5.0842
380	1.5361	642.72	5.1019
390	1.4963	649.44	5.1194
400	1.4585	656.23	5.1365
410	1.4226	663.08	5.1535
420	1.3885	670.00	5.1701
430	1.3559	676.98	5.1865
440	1.3249	684.01	5.2027
450	1.2952	691.10	5.2187
460	1.2669	698.25	5.2344
470	1.2398	705.45	5.2498
480	1.2138	712.70	5.2651
490	1.1889	720.00	5.2802
500	1.1650	727.34	5.2950

Temp K	Abs Pressure 0.101325 MPa (Sat'n Temp 243.36 K)		
t	1/v	h	s
(Sat'n)	6.3048	557.33	4.7618
250	6.1128	561.16	4.7773
260	5.8483	566.96	4.8001
270	5.6089	572.83	4.8222
280	5.3906	578.77	4.8438
290	5.1903	584.78	4.8649
300	5.0056	590.87	4.8856
310	4.8346	597.04	4.9058
320	4.6756	603.28	4.9256
330	4.5272	609.60	4.9450
340	4.3885	615.99	4.9641
350	4.2584	622.46	4.9829
360	4.1360	629.00	5.0013
370	4.0207	635.61	5.0194
380	3.9118	642.28	5.0372
390	3.8088	649.02	5.0547
400	3.7112	655.83	5.0719
410	3.6186	662.70	5.0889
420	3.5306	669.63	5.1056
430	3.4469	676.63	5.1221
440	3.3670	683.67	5.1383
450	3.2909	690.78	5.1542
460	3.2181	697.93	5.1700
470	3.1486	705.14	5.1855
480	3.0820	712.40	5.2007
490	3.0182	719.71	5.2158
500	2.9570	727.06	5.2307

Temp K	Abs Pressure 0.20 MPa (Sat'n Temp 260.60 K)		
t	1/v	h	s
(Sat'n)	11.907	565.28	4.7490
270	11.397	571.03	4.7706
280	10.912	577.17	4.7930
290	10.473	583.35	4.8146
300	10.075	589.58	4.8358
310	9.709	595.86	4.8564
320	9.373	602.20	4.8765
330	9.062	608.61	4.8962
340	8.772	615.07	4.9155
350	8.502	621.60	4.9344
360	8.249	628.19	4.9530
370	8.012	634.85	4.9712
380	7.789	641.57	4.9891
390	7.578	648.35	5.0067
400	7.379	655.19	5.0241
410	7.191	662.09	5.0411
420	7.012	669.04	5.0579
430	6.843	676.06	5.0744
440	6.681	683.13	5.0906
450	6.528	690.25	5.1066
460	6.381	697.42	5.1224
470	6.241	704.65	5.1379
480	6.107	711.92	5.1533
490	5.979	719.25	5.1683
500	5.856	726.61	5.1832

Temp K	Abs Pressure 0.40 MPa (Sat'n Temp 281.28 K)		
t	1/v	h	s
(Sat'n)	22.978	574.47	4.7390
290	22.018	580.21	4.7591
300	21.042	586.78	4.7814
310	20.173	593.33	4.8029
320	19.389	599.90	4.8237
330	18.676	606.50	4.8440
340	18.023	613.13	4.8638
350	17.422	619.79	4.8832
360	16.865	626.51	4.9021
370	16.347	633.27	4.9206
380	15.864	640.08	4.9387
390	15.411	646.94	4.9566
400	14.986	653.85	4.9741
410	14.585	660.81	4.9913
420	14.206	667.83	5.0082
430	13.849	674.89	5.0248
440	13.509	682.01	5.0411
450	13.187	689.17	5.0572
460	12.881	696.38	5.0731
470	12.589	703.64	5.0887
480	12.311	710.95	5.1041
490	12.045	718.30	5.1193
500	11.791	725.70	5.1342

Temp K	Abs Pressure 0.60 MPa (Sat'n Temp 295.15 K)		
t	1/v	h	s
(Sat'n)	34.045	580.27	4.7345
300	33.182	583.66	4.7459
310	31.587	590.56	4.7685
320	30.190	597.42	4.7903
330	28.949	604.24	4.8113
340	27.834	611.06	4.8316
350	26.821	617.89	4.8514
360	25.895	624.74	4.8707
370	25.043	631.62	4.8896
380	24.254	638.53	4.9080
390	23.521	645.48	4.9261
400	22.837	652.47	4.9438
410	22.196	659.51	4.9611
420	21.594	666.58	4.9782
430	21.027	673.70	4.9949
440	20.491	680.87	5.0114
450	19.985	688.08	5.0276
460	19.505	695.33	5.0436
470	19.049	702.63	5.0592
480	18.615	709.97	5.0747
490	18.201	717.35	5.0899
500	17.807	724.78	5.1049

Temp K	Abs Pressure 0.80 MPa (Sat'n Temp 305.91 K)		
t	1/v	h	s
(Sat'n)	45.295	584.49	4.7316
310	44.253	587.48	4.7414
320	41.983	594.70	4.7643
330	40.026	601.81	4.7861
340	38.307	608.86	4.8072
350	36.776	615.88	4.8276
360	35.396	622.89	4.8473
370	34.141	629.91	4.8665
380	32.993	636.94	4.8853
390	31.934	643.98	4.9036
400	30.953	651.06	4.9215
410	30.040	658.17	4.9390
420	29.187	665.31	4.9563
430	28.387	672.49	4.9731
440	27.636	679.71	4.9897
450	26.927	686.97	5.0060
460	26.258	694.26	5.0221
470	25.624	701.60	5.0379
480	25.022	708.98	5.0534
490	24.450	716.39	5.0687
500	23.906	723.85	5.0837

Temp K	Abs Pressure 1.00 MPa (Sat'n Temp 314.85 K)		
t	1/v	h	s
(Sat'n)	56.833	587.74	4.7294
320	55.077	591.68	4.7418
330	52.110	599.16	4.7648
340	49.583	606.50	4.7868
350	47.385	613.75	4.8078
360	45.439	620.95	4.8281
370	43.696	628.11	4.8477
380	42.119	635.27	4.8668
390	40.680	642.43	4.8854
400	39.357	649.61	4.9035
410	38.135	656.80	4.9213
420	37.000	664.01	4.9387
430	35.941	671.26	4.9557
440	34.951	678.53	4.9725
450	34.021	685.84	4.9889
460	33.145	693.18	5.0050
470	32.319	700.56	5.0209
480	31.537	707.98	5.0365
490	30.795	715.43	5.0519
500	30.091	722.91	5.0670

Courtesy of ASHRAE

REFRIGERANT NUMBERS
TABLE 3

Refrigerant Number	Chemical Name	Chemical Formula
Halocarbon Compounds		
10	Carbontetrachloride	CCl_4
11	Trichlorofluoromethane	CCl_3F
12	Dichlorodifluoromethane	CCl_2F_2
13	Chlorotrifluoromethane	$CClF_3$
13B1	Bromotrifluoromethane	$CBrF_3$
14	Carbontetrafluoride	CF_4
20	Chloroform	$CHCl_3$
21	Dichlorofluoromethane	$CHCl_2F$
22	Chlorodifluoromethane	$CHClF_2$
23	Trifluoromethane	CHF_3
30	Methylene Chloride	CH_2Cl_2
31	Chlorofluoromethane	CH_2ClF
32	Methylene Fluoride	CH_2F_2
40	Methyl Chloride	CH_3Cl
41	Methyl Fluoride	CH_3F
50*	Methane	CH_4
110	Hexachloroethane	CCl_3CCl_3
111	Pentachlorofluoroethane	CCl_3CCl_2F
112	Tetrachlorodifluoroethane	CCl_2FCCl_2F
112a	Tetrachlorodifluoroethane	CCl_3CClF_2
113	Trichlorotrifluoroethane	CCl_2FCClF_2
113a	Trichlorotrifluoroethane	CCl_3CF_3
114	Dichlorotetrafluoroethane	$CClF_2CClF_2$
114a	Dichlorotetrafluoroethane	CCl_2FCF_3
114B2	Dibromotetrafluoroethane	$CBrF_2CBrF_2$
115	Chloropentafluoroethane	$CClF_2CF_3$
116	Hexafluoroethane	CF_3CF_3
120	Pentachloroethane	$CHCl_2CCl_3$
123	Dichlorotrifluoroethane	$CHCl_2CF_3$
124	Chlorotetrafluoroethane	$CHClFCF_3$
124a	Chlorotetrafluoroethane	CHF_2CClF_2
125	Pentafluoroethane	CHF_2CF_3
133a	Chlorotrifluoroethane	CH_2ClCF_3
140a	Trichloroethane	CH_3CCl_3
142b	Chlorodifluoroethane	CH_3CClF_2
143a	Trifluoroethane	CH_3CF_3
150a	Dichloroethane	CH_3CHCl_2
152a	Difluoroethane	CH_3CHF_2
160	Ethyl Chloride	CH_3CH_2Cl
170*	Ethane	CH_3CH_3
218	Octafluoropropane	$CF_3CF_2CF_3$
290*	Propane	$CH_3CH_2CH_3$
Cyclic Organic Compounds		
C316	Dichlorohexafluorocyclobutane	$C_4Cl_2F_6$
C317	Chloroheptafluorocyclobutane	C_4ClF_7
C318	Octafluorocyclobutane	C_4F_8

Refrigerant Number	Chemical Name	Chemical Formula
Azeotropes		
500	Refrigerants 12/152a	CCl_2F_2/CH_3CHF_2
501	Refrigerants 22/12	$CHClF_2/CCl_2F_2$
502	Refrigerants 22/115	$CHClF_2/CClF_2CF_3$
503	Refrigerants 23/13	$CHF_3/CClF_3$
504	Refrigerants 32/115	$CH_2F_2/CClF_2CF_3$
505	Refrigerants 12/31	CCl_2F_2/CH_2ClF
506	Refrigerants 31/114	$CH_2ClF/CClF_2CClF_2$
Miscellaneous Organic Compounds		
Hydrocarbons		
50	Methane	CH_4
170	Ethane	CH_3CH_3
290	Propane	$CH_3CH_2CH_3$
600	Butane	$CH_3CH_2CH_2CH_3$
600a	Isobutane (2 methyl propane)	$CH(CH_3)_3$
1150**	Ethylene	$CH_2{=}CH_2$
1270**	Propylene	$CH_3CH{=}CH_2$
Oxygen Compounds		
610	Ethyl Ether	$C_2H_5OC_2H_5$
611	Methyl Formate	$HCOOCH_3$
Nitrogen Compounds		
630	Methyl Amine	CH_3NH_2
631	Ethyl Amine	$C_2H_5NH_2$
Inorganic Compounds		
702	Hydrogen (Normal and Para)	H_2
704	Helium	He
717	Ammonia	NH_3
718	Water	H_2O
720	Neon	Ne
728	Nitrogen	N_2
729	Air	$.21O_2$, $.78N_2$, .01A
732	Oxygen	O_2
740	Argon	A
744	Carbon Dioxide	CO_2
744A	Nitrous Oxide	N_2O
764	Sulfur Dioxide	SO_2
Unsaturated Organic Compounds		
1112a	Dichlorodifluoroethylene	$CCl_2{=}CF_2$
1113	Chlorotrifluoroethylene	$CClF{=}CF_2$
1114	Tetrafluoroethylene	$CF_2{=}CF_2$
1120	Trichloroethylene	$CHCl{=}CCl_2$
1130	Dichloroethylene	$CHCl{=}CHCl$
1132a	Vinylidene Fluoride	$CH_2{=}CF_2$
1140	Vinyl Chloride	$CH_2{=}CHCl$
1141	Vinyl Fluoride	$CH_2{=}CHF$
1150	Ethylene	$CH_2{=}CH_2$
1270	Propylene	$CH_3CH{=}CH_2$

*Methane, ethane, and propane appear in the Halocarbon section in their proper numerical order, but these compounds are not halocarbons.

**Ethylene and propylene appear in the Hydrocarbon section to indicate that these compounds are hydrocarbons, but are properly identified in the section Unsaturated Organic Compounds.

Courtesy of ASHRAE

PHYSICAL PROPERTIES OF SOME REFRIGERANTS IN SI UNITS
TABLE 4

Refrigerant No.	Refrigerant Name	Chemical Formula	Molecular Mass	Boiling Point (NBP),[f] °C	Freezing Point, °C	Critical Temperature, °C	Critical Pressure, kPa	Critical Volume, L/kg
704	Helium	He	4.0026	−268.9	None	−267.9	228.8	14.43
702n	Hydrogen	H_2	2.0159	−252.8	−259.2	−239.9	1315	33.21
729	Air	. . .	28.97	−194.3	. . .	−140.7	3772	3.048
						−140.6	3764	3.126
732	Oxygen	O_2	31.9988	−182.9	−218.8	−118.4	5077	2.341
50	Methane	CH_4	16.04	−161.5	−182.2	− 82.5	4638	6.181
14	Tetrafluoromethane	CF_4	88.01	−127.9	−184.9	− 45.7	3741	1.598
1150	Ethylene	C_2H_4	28.05	−103.7	−169	9.3	5114	4.37
503	b	. . .	87.5	− 88.7	. . .	19.5	4182	2.035
23	Trifluoromethane	CHF_3	70.02	− 82.1	−155	25.6	4833	1.942
13	Chlorotrifluoromethane	$CClF_3$	104.47	− 81.4	−181	28.8	3865	1.729
744	Carbon Dioxide	CO_2	44.01	− 78.4	− 56.6[a]	31.1	7372	2.135
504	c	. . .	79.2	− 57.2	. . .	66.4	4758	2.023
1270	Propylene	C_3H_6	42.09	− 47.7	−185	91.8	4618	4.495
502	d	. . .	111.63	− 45.4	. . .	82.2	4072	1.785
290	Propane	C_3H_8	44.10	− 42.07	−187.7	96.8	4254	4.545
22	Chlorodifluoromethane	$CHClF_2$	86.48	− 40.76	−160	96.0	4974	1.904
115	Chloropentafluoroethane	$CClF_2CF_3$	154.48	− 39.1	−106	79.9	3153	1.629
717	Ammonia	NH_3	17.03	− 33.3	− 77.7	133.0	11417	4.245
500	e	. . .	99.31	− 33.5	−159	105.5	4423	2.016
12	Dichlorodifluoromethane	CCl_2F_2	120.93	− 29.79	−158	112.0	4113	1.792
152	Difluoroethane	CH_3CHF_2	66.05	− 25.0	−117	113.5	4492	2.741
764	Sulfur Dioxide	SO_2	64.07	− 10.0	− 75.5	157.5	7875	1.910
142b	Chlorodifluoroethane	CH_3CClF_2	100.5	− 9.8	−131	137.1	4120	2.297
630	Methyl Amine	CH_3NH_2	31.06	− 6.7	− 92.5	156.9	7455	
C318	Octafluorocyclobutane	C_4F_8	200.04	− 5.8	− 41.4	115.3	2781	1.611
600	Butane	C_4H_{10}	58.13	− 0.5	−138.5	152.0	3794	4.383
21	Dichlorofluoromethane	$CHCl_2F$	102.92	8.8	−135	178.5	5168	1.917
631	Ethyl Amine	$C_2H_5NH_2$	45.08	16.6	− 80.6	183.0	5619	
11	Trichlorofluoromethane	CCl_3F	137.38	23.82	−111	198.0	4406	1.804
610	Ethyl Ether	$C_4H_{10}O$	74.12	34.6	−116.3	194.0	3603	3.790
216	Dichlorohexafluoropropane	$C_3Cl_2F_6$	220.93	35.69	−125.4	180.0	2753	1.742
1120	Trichloroethylene	$CHCl{=}CCl_2$	131.39	87.2	− 73	271.1	5016	
718	Water	H_2O	18.02	100	0	374.2	22103	3.128

[a] At 527 kPa.
[b] Refrigerants 23 and 13 (40.1/59.9% by mass).
[c] Refrigerants 32 and 115 (48.2/51.8% by mass).
[d] Refrigerants 22 and 115 (48.8/51.2% by mass).
[e] Refrigerants 12 and 152a (73.8/26.2% by mass).
[f] At normal atmospheric pressure (101.3 kPa).

Courtesy of ASHRAE

UNDERWRITERS' LABORATORIES CLASSIFICATION OF COMPARATIVE HAZARD TO LIFE OF GASES AND VAPORS
TABLE 5

Refrigerant No.	Refrigerant Name	ANSI B9.1 - 1971 Safety Code Group	Underwriters' Laboratories Group Classification	Explosive Limits in Air, % by Volume
50	Methane	3[a]	5b	4.9 to 15.0
14	Tetrafluoromethane	1[a]	6[a]	Nonflammable
1150	Ethylene	3[a]	5b	3.0 to 25.0
13	Chlorotrifluoromethane	1	6	Nonflammable
744	Carbon Dioxide	1	5*	Nonflammable
13B1	Bromotrifluoromethane	1[a]	6	Nonflammable
290	Propane	3	5b*	2.3 to 7.3
502	. . .	1	5a	Nonflammable
22	Chlorodifluoromethane	1	5a**	Nonflammable
717	Ammonia	2	2*	16.0 to 25.0
500	. . .	1	5a	Nonflammable
12	Dichlorodifluoromethane	1	6*	Nonflammable
764	Sulfur Dioxide	2	1*	Nonflammable
600	Butane	3	5*	1.6 to 6.5
21	Dichlorofluoromethane	1	4-5†	Nonflammable
11	Trichlorofluoromethane	1	5*	Nonflammable

*Underwriters' Laboratories Report MH-2375.
**Underwriters' Laboratories Report MH-3134.
†Underwriters' Laboratories Report MH-2630.

Group	Definition	Examples
1	Gases or vapors which in concentrations of about ½ to 1 percent for durations of exposure of about 5 minutes are lethal or produce serious injury.	Sulfur Dioxide
2	Gases or vapors which in concentrations of about ½ to 1 percent for durations of exposure of about ½ hour are lethal or produce serious injury.	Ammonia Methyl Bromide
3	Gases or vapors which in concentration of about 2 to 2½ percent for durations of exposure of about 1 hour are lethal or produce serious injury.	Carbon Tetrachloride Chloroform Methyl Formate
4	Gases or vapors which in concentrations of about 2 to 2½ percent for durations of exposure of about 2 hours are lethal or produce serious injury.	Dichloroethylene Methyl Chloride Ethyl Bromide
Between 4 & 5	Appear to classify as somewhat less toxic than Group 4.	Methylene Chloride Ethyl Chloride
	Much less toxic than Group 4 but somewhat more toxic than Group 5.	Refrigerant 113
5a	Gases or vapors much less toxic than Group 4 but more toxic than Group 6.	Refrigerant 11 Refrigerant 22 Carbon Dioxide
5b	Gases or vapors which available data indicate would classify as either Group 5a or Group 6.	Ethane Propane Butane
6	Gases or vapors which in concentrations up to at least about 20 percent by volume for durations of exposure of about 2 hours do not appear to produce injury.	Refrigerant 12 Refrigerant 114 Refrigerant 13B1

COMPARISON BETWEEN FOUR COMMON TYPES OF REFRIGERATION COMPRESSORS
TABLE 6

Advantages	Disadvantages
Rotary and Vane	
• Good efficiency as booster:equal to screw and better than piston type • Handles low pressure conditions • Mechanically reliable	• Discharge pressure limitation • Overall pressure ratio limited to about 7:1 • Poor part load power characteristics
Reciprocating Piston	
• Basic industry work horse • Full range of sizes & capacities • Efficient part load operation • Relatively inexpensive • Requires minimum amount of support infra-structure	• Volumetric efficiency drops at high overall pressure ratios • Requires frequent maintenance • Not tolerant of liquid • Water cooling necessary for ammonia systems
Rotary Screw	
• Good efficiency at full load • Large capacity units available • Low maintenance costs • Reliable • Tolerant to liquid • Liquid injection cooling option • Infinitely variable capacity control • High operating flexibility	• Poor power performance at low part load conditions • Small sizes expensive • Repairs expensive in remote locations
Centrifugal	
• Efficient at full load • Large capacity units require small space	• Very high speed precision equipment • Useable only with freon type refrigerants • Inefficient at part load • Severe operating restrictions

THE EFFECTS OF POOR MAINTENANCE ON THE EFFICIENCY OF A 17.5 KW CAPACITY RECIPROCATING COMPRESSOR
TABLE 7A

Condition	Suction Temperature (°C)	Discharge Temperature (°C)	Compressor Capacity (kW)	Power Input (kW)	Cooling kW per Input kW	Per Cent Capacity Reduction
Normal	4.4	48.9	18.2	6.5	2.8	—
Dirty Evaporator	2.8	48.9	17.2	6.3	2.7	5.6
Dirty Condenser	5.6	54.4	17.3	7.1	2.4	4.8
Dirty Evaporator and Condenser	1.7	54.4	14.6	6.5	2.3	19.4

THE EFFECTS OF POOR MAINTENANCE ON THE EFFICIENCY OF A 60 KW CAPACITY RECIPROCATING COMPRESSOR
TABLE 7B

Condition	Suction Temperature (°C)	Discharge Temperature (°C)	Compressor Capacity (kW)	Power Input (kW)	Cooling kW per Input kW	Per Cent Capacity Reduction
Normal	7.2	40.6	59.8	11.9	5.0	—
Dirty Evaporator	1.7	40.1	48.5	11.4	4.3	18.9
Dirty Condenser	7.2	46.1	54.9	13.1	4.2	8.2
Dirty Evaporator and Condenser	1.7	46.1	44.7	12.2	3.7	25.4

THE EFFECTS OF POOR MAINTENANCE ON THE EFFICIENCY OF A 1830 KW ABSORPTION CHILLER
TABLE 8

Condition	Chilled Water Temperature (°C)	Condenser Water Temperature (°C)	Cooling Capacity (kW)	Steam Input (kg/kw.h)	Per Cent Increase In Steam Input	Per Cent Capacity Reduction
Normal	6.7	29.4	1029	2.42	—	—
Dirty Evaporator	4.4	32.2	1646	2.48	2.5	10
Dirty Condenser	6.7	29.4	1607	2.49	3	12
Dirty Evaporator and Condenser	4.4	32.2	1393	2.60	7.5	23.8

COMMON CONVERSIONS

1 barrel (35 Imp gal) (42 US gal)	= 159.1 litres	1 kilowatt	= 3600 kilojoules
1 gallon (Imp)	= 1.20094 gallon (US)	1 Newton	= 1 kg-m/s^2
1 horsepower (boiler)	= 9809.6 watts	1 therm	= 10^5 Btu
1 horsepower	= 2545 Btu/hour	1 ton (refrigerant)	= 12002.84 Btu/hour
1 horsepower	= 0.746 kilowatts	1 ton (refrigerant)	= 3516.8 watts
1 joule	= 1 N-m	1 watt	= 1 joule/second
Kelvin	= (°C + 273.15)	Rankine	= (°F + 459.67)

Cubes

1 yd^3	=	27 ft^3
1 ft^3	=	1728 in^3
1 cm^3	=	1000 mm^3
1 m^3	=	10^6 cm^3
1 m^3	=	1000 L

Squares

1 yd^2	=	9 ft^2
1 ft^2	=	144 in^2
1 cm^2	=	100 mm^2
1 m^2	=	10000 cm^2

SI PREFIXES

Prefix	Symbol	Magnitude	Factor
tera	T	1 000 000 000 000	10^{12}
giga	G	1 000 000 000	10^9
mega	M	1 000 000	10^6
kilo	k	1 000	10^3
hecto	h	100	10^2
deca	da	10	10^1
deci	d	0.1	10^{-1}
centi	c	0.01	10^{-2}
milli	m	0.001	10^{-3}
micro	u	0.000 001	10^{-6}
nano	n	0.000 000 001	10^{-9}
pica	p	0.000 000 000 001	10^{-12}

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UNIT CONVERSION TABLES
METRIC TO IMPERIAL

FROM	SYMBOL	TO	SYMBOL	MULTIPLY BY
amperes/square centimetre	A/cm^2	amperes/square inch	A/in^2	6.452
Celsius	°C	Fahrenheit	°F	(°C × 9/5) + 32
centimetres	cm	inches	in	0.3937
cubic centimetres	cm^3	cubic inches	in^3	0.06102
cubic metres	m^3	cubic foot	ft^3	35.314
grams	g	ounces	oz	0.03527
grams	g	pounds	lb	0.0022
grams/litre	g/L	pounds/cubic foot	lb/ft^3	0.06243
joules	J	Btu	Btu	9.480×10^{-4}
joules	J	foot-pounds	ft-lb	0.7376
joules	J	horsepower-hours	hp-h	3.73×10^{-7}
joules/metre, (Newtons)	J/m, N	pounds	lb	0.2248
kilograms	kg	pounds	lb	2.205
kilograms	kg	tons (long)	ton	9.842×10^{-4}
kilograms	kg	tons (short)	tn	1.102×10^{-3}
kilometres	km	miles (statute)	mi	0.6214
kilopascals	kPa	atmospheres	atm	9.87×10^{-3}
kilopascals	kPa	inches of mercury (@ 32°F)	in Hg	0.2953
kilopascals	kPa	inches of water (@ 4°C)	in H_2O	4.0147
kilopascals	kPa	pounds/square inch	psi	0.1450
kilowatts	kW	foot-pounds/second	ft-lb/s	737.6
kilowatts	kW	horsepower	hp	1.341
kilowatt-hours	kWh	Btu	Btu	3413
litres	L	cubic foot	ft^3	0.03531
litres	L	gallons (Imp)	gal (Imp)	0.21998
litres	L	gallons (US)	gal (US)	0.2642
litres/second	L/s	cubic foot/minute	cfm	2.1186
lumen/square metre	lm/m^2	lumen/square foot	lm/ft^2	0.09290
lux, lumen/square metre	lx, lm/m^2	footcandles	fc	0.09290
metres	m	foot	ft	3.281
metres	m	yard	yd	1.09361
parts per million	ppm	grains/gallon (Imp)	gr/gal (Imp)	0.07
parts per million	ppm	grains/gallon (US)	gr/gal (US)	0.05842
permeance (metric)	PERM	permeance (Imp)	perm	0.01748
square centimetres	cm^2	square inches	in^2	0.1550
square metres	m^2	square foot	ft^2	10.764
square metres	m^2	square yards	yd^2	1.196
tonne (metric)	t	pounds	lb	2204.6
watt	W	Btu/hour	Btu/h	3.413
watt	W	lumen	lm	668.45

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UNIT CONVERSION TABLES
IMPERIAL TO METRIC

FROM	SYMBOL	TO	SYMBOL	MULTIPLY BY
ampere/in^2	A/in^2	ampere/cm^2	A/cm^2	0.1550
atmospheres	atm	kilopascals	kPa	101.325
British Thermal Unit	Btu	joules	J	1054.8
Btu	Btu	kilogram-metre	kg-m	107.56
Btu	Btu	kilowatt-hour	kWh	2.928×10^{-4}
Btu/hour	Btu/h	watt	W	0.2931
calorie, gram	cal or g-cal	joules	J	4.186
chain	chain	metre	m	20.11684
cubic foot	ft^3	cubic metre	m^3	0.02832
cubic foot	ft^3	litre	L	28.32
cubic foot/minute	cfm	litre/second	L/s	0.47195
cycle/second	c/s	Hertz	Hz	1.00
Fahrenheit	°F	Celsius	°C	(°F-32)/1.8
foot	ft	metre	m	0.3048
footcandle	fc	lux, lumen/ square metre	lx, lm/m^2	10.764
footlambert	fL	candela/square metre	cd/m^2	3.42626
foot-pounds	ft-lb	joule	J	1.356
foot-pounds	ft-lb	kilogram-metres	kg-m	0.1383
foot-pounds/second	ft-lb/s	kilowatt	kW	1.356×10^{-3}
gallons (Imp)	gal (Imp)	litres	L	4.546
gallons (US)	gal (US)	litres	L	3.785
grains/gallon (Imp)	gr/gal (Imp)	parts per million	ppm	14.286
grains/gallon (US)	gr/gal (US)	parts per million	ppm	17.118
horsepower	hp	watts	W	745.7
horsepower-hours	hp-h	joules	J	2.684×10^6
inches	in	centimetres	cm	2.540
inches of Mercury (@ 32°F)	in Hg	kilopascals	kPa	3.386
inches of water (@ 4°C)	in H_2O	kilopascals	kPa	0.2491

UNIT CONVERSION TABLES
IMPERIAL TO METRIC (cont'd)

FROM	SYMBOL	TO	SYMBOL	MULTIPLY BY
lamberts	* L	candela/square metre	cd/m^2	3.183
lumen/square foot	lm/ft^2	lumen/square metre	lm/m^2	10.76
lumen	lm	watt	W	0.001496
miles (statute)	mi	kilometres	km	1.6093
ounces	oz	grams	g	28.35
perm (at 0°C)	perm	kilogram per pascal-second-square metre	$kg/Pa\text{-}s\text{-}m^2$ (PERM)	5.721×10^{-11}
perm (at 23°C)	perm	kilogram per pascal-second-square metre	$kg/Pa\text{-}s\text{-}m^2$ (PERM)	5.745×10^{-11}
perm-inch (at 0°C)	perm. in.	kilogram per pascal-second-metre	kg/Pa-s-m	1.4532×10^{-12}
perm-inch (at 23°C)	perm. in.	kilogram per pascal-second-metre	kg/Pa-s-m	1.4593×10^{-12}
pint (Imp)	pt	litre	L	0.56826
pounds	lb	grams	g	453.5924
pounds	lb	joules/metre, (Newtons)	J/m, N	4.448
pounds	lb	kilograms	kg	0.4536
pounds	lb	tonne (metric)	t	4.536×10^{-4}
pounds/cubic foot	lb/ft^3	grams/litre	g/L	16.02
pounds/square inch	psi	kilopascals	kPa	6.89476
quarts	qt	litres	L	1.1365
slug	slug	kilograms	kg	14.5939
square foot	ft^2	square metre	m^2	0.09290
square inches	in^2	square centimetres	cm^2	6.452
square yards	yd^2	square metres	m^2	0.83613
tons (long)	ton	kilograms	kg	1016
tons (short)	tn	kilograms	kg	907.185
yards	yd	metres	m	0.9144

* "L" as used in Lighting

4138T/T33(-2)

The following typical values for conversion factors may be used when actual data are unavailable. The MJ and Btu equivalencies are heats of combustion. Hydrocarbons are shown at the higher heating value, wet basis. Some items listed are typically feedstocks, but are included for completeness and as a reference source. The conversion factors for coal are approximate since the heating value of a specific coal is dependent on the particular mine from which it is obtained.

Consistent factors must be used when calculating Base Year and Current Year energy usage.

ENERGY TYPE	METRIC	IMPERIAL
COAL		
— metallurgical	29,000 megajoules/tonne	25.0×10^6 Btu/ton
— anthracite	30,000 megajoules/tonne	25.8×10^6 Btu/ton
— bituminous	32,100 megajoules/tonne	27.6×10^6 Btu/ton
— sub-bituminous	22,100 megajoules/tonne	19.0×10^6 Btu/ton
— lignite	16,700 megajoules/tonne	14.4×10^6 Btu/ton
COKE		
— metallurgical	30,200 megajoules/tonne	26.0×10^6 Btu/ton
— petroleum		
— raw	23,300 megajoules/tonne	20.0×10^6 Btu/ton
— calcined	32,600 megajoules/tonne	28.0×10^6 Btu/ton
PITCH	37,200 megajoules/tonne	32.0×10^6 Btu/ton
CRUDE OIL	38,5 megajoules/litre	5.8×10^6 Btu/bbl
No. 2 OIL	38.68 megajoules/litre	5.88×10^6 Btu/bbl $.168 \times 10^6$ Btu/IG
No. 4 OIL	40.1 megajoules/litre	6.04×10^6 Btu/bbl $.173 \times 10^6$ Btu/IG
No. 6 OIL (RESID. BUNKER C)		
@ 2.5% sulphur	42.3 megajoules/litre	6.38×10^6 Btu/bbl $.182 \times 10^6$ Btu/IG
@ 1.0% sulphur	40.5 megajoules/litre	6.11×10^6 Btu/bbl $.174 \times 10^6$ Btu/IG
@ .5% sulphur	40.2 megajoules/litre	6.05×10^6 Btu/bbl $.173 \times 10^6$ Btu/IG
KEROSENE	37.68 megajoules/litre	$.167 \times 10^6$ Btu/IG
DIESEL FUEL	38.68 megajoules/litre	$.172 \times 10^6$ Btu/IG
GASOLINE	36.2 megajoules/litre	$.156 \times 10^6$ Btu/IG
NATURAL GAS	37.2 megajoules/m^3	1.00×10^6 Btu/MCF
PROPANE	50.3 megajoules/kg 26.6 megajoules/litre	$.02165 \times 10^6$ Btu/lb $.1145 \times 10^6$ Btu/IG
ELECTRICITY	3.6 megajoules/kWh	$.003413 \times 10^6$ Btu/kWh

Energy Costs
Worksheet 11-1
Page 1 of 2

Company: ______________________ Date: ______________________

Location: ______________________ By: ______________________

Energy Used or Displaced Per Year = ______________________ MJ/yr

Details:

$$\text{or,} = \frac{\text{\_\_\_\_\_\_\_\_\_\_\_\_\_\_ MJ/yr}}{3.6 \text{ MJ/kWh}}$$

= ______________________ kWh/yr (1)

Fuel Costs:

- Gas . . . . ($/m³) ______________(2)
- Oil . . . . ($/L) ______________(3)
- Electricity . . . . ($/kWh) ______________ (4) (Ignore demand charges)
- Other . . . . ($/unit) ______________ (5) ______________ unit (L,m³,tonne,kg)

Energy content of fuels (other than electricity)

- Fuel type ______________________
- Energy content ______________ MJ per ______________________ (6)
  (L,kg,m³,tonne)

(From plant data or Appendix C)

Energy Costs
Worksheet 11-1

Company: ______________________ Date: ______________________

Location: ______________________ By: ______________________

Cost of energy used or displaced per year

- Gas

$$AG = \frac{\_\_\_\_\_\_\_\_ (1) \times \_\_\_\_\_\_\_\_ (2)}{\_\_\_\_\_\_\_\_ (6) \times \_\_\_\_\_\_\_\_ (\text{eff.})} = \$\_\_\_\_\_\_\_\_ /\text{yr}$$

- Gas

$$AG = \frac{\_\_\_\_\_\_\_\_ (1) \times \_\_\_\_\_\_\_\_ (2)}{\_\_\_\_\_\_\_\_ (6) \times \_\_\_\_\_\_\_\_ (\text{eff.})} = \$\_\_\_\_\_\_\_\_ /\text{yr}$$

- Oil

$$AO = \frac{\_\_\_\_\_\_\_\_ (1) \times \_\_\_\_\_\_\_\_ (3)}{\_\_\_\_\_\_\_\_ (6) \times \_\_\_\_\_\_\_\_ (\text{eff.})} = \$\_\_\_\_\_\_\_\_ /\text{yr}$$

- Electricity

$$AE = \_\_\_\_\_\_\_\_ (1) \times \_\_\_\_\_\_\_\_ (4) = \$\_\_\_\_\_\_\_\_ /\text{yr}$$

- Other

$$AF = \frac{\_\_\_\_\_\_\_\_ (1) \times \_\_\_\_\_\_\_\_ (5)}{\_\_\_\_\_\_\_\_ (6) \times \_\_\_\_\_\_\_\_ (\text{eff.})} = \$\_\_\_\_\_\_\_\_ /\text{yr}$$

Where, eff. = efficiency of heating system

AG = annual cost of gas

AO = annual cost of oil

AE = annual cost of electricity

AF = annual cost of fuel

For gas and oil fired systems,eff. = 0.8

For electric systems, eff. = 1.0

Heat Energy Available From Waste Water Stream
To Preheat A Water Stream
(Approximate Method)
Worksheet 11-2
Page 1 of 2

Company: ______________________ Date: ______________________

Location: ______________________ By: ______________________

Waste Water Stream ____

- Water flow (fw) ______________________ L/s
- Present water temperature (t_1) ______________________ °C
- Proposed water leaving temperature (t_2) ______________________ °C
  (Input of heat exchanger manufacturer required to establish this figure, or estimate using $t_2 = t_1 - 5°$)
- Heat available, (Q) $= fw \times (t_1 - t_2) \times 15$ MJ/h

$= \_\_\_\_\_\_\_\_ \times (\_\_\_\_\_\_\_\_ - \_\_\_\_\_\_\_\_) \times 15$

$= \_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_$ MJ/h

or $\dfrac{\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_ \text{ MJ/h}}{3600 \text{ s/h}}$

$= \_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_$ MJ/s

Proposed heat exchanger efficiency ____________%
(from heat exchanger manufacturer)

Heat available $= \_\_\_\_\_\_\_\_\_\_\_\_$ MJ/s $\times \dfrac{\_\_\_\_\_\_\_\_ \%}{100}$

$= \_\_\_\_\_\_\_\_\_\_\_\_$ MJ/s (1)

Heat Energy Available From Waste Water Stream To Preheat A Water Stream

(Approximate Method)

Worksheet 11-2

Page 2 of 2

Company: ______________________ Date: ______________________

Location: ______________________ By: ______________________

Process Stream

Water flow (fw) ______________ L/s

Entering water temperature (t_1) ______________ °C

Required water temperature (t_2) ______________ °C

Heat required, (Q) = fw x (t_1 − t_2) x 15MJ/h

= __________ x (__________ − __________) x 15

= − ______________________ MJ/h

or $\dfrac{\text{\_\_\_\_\_\_\_\_\_\_\_\_\_\_ MJ/h}}{3600 \text{ s/h}}$ (drop negative sign)

= ______________________ MJ/s (2)

Heat energy required for final heating of process water stream

= __________ (2) − __________ (1)

= ______________________ MJ/s (3)